THE ELEMENTS OF INTERNAL COMBUSTION ENGINEERING

TELFORD PETRIE

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THE ELEMENTS OF INTERNAL-COMBUSTION ENGINEERING

BY

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WITH 62 *DIAGRAMS*

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PREFACE

No one, I think, will deny the important position which the internal-combustion engine has come to occupy in the family of prime-movers. It is true that, up to the present, it has inspired nothing so wonderful as "M'Andrews' Hymn," but one hopes that the time is not so far distant when the engineer's Poet Laureate will sing its praises in "A Ballad of Oil." The days are gone by when the reciprocating steam engine can be allowed to dominate the heat-engine papers of the more advanced examinations. Although it is by far the oldest member of the family, it is no longer the strongest, nor even the most important, and those sturdy twins, brother turbine and brother gas engine, both now some forty years old, are clamouring for at least an equal share of questions. It is with the hope of helping one of these flourishing upstarts to obtain its fair share of recognition that this book has been written. It deals almost entirely with what may be termed the "paper" side of engineering, and leaves all practical details to those many excellent books on design which it would be invidious to detail.

The thermodynamics of the subject must, of necessity, owe their chief inspiration to Sir Dugald Clerk and his coadjutor, Mr. G. A. Burls; but, in my case, the work of a former chief, Mr. H. E. Wimperis, and of two erstwhile colleagues, Mr. S. Lees and Mr. W. J. Walker, has also had no little influence. One of my present colleagues, Mr. T. Bevan, by collecting, arranging, and working out all the examples, which will be found at the end, has added considerably to the primary object of the book. The task was made easy by the courtesy of the authorities at the Universities of Cambridge, London, and Manchester, who granted permission for the reproduction of examples from their various Engineering Degree papers. Nor must I forget to mention Mr. C. Cooper, who has corrected all the proofs, and a senior student here, Mr. H. Knowles, who has rendered much help in checking, in one case inspiring, some of the more original portions of the book.

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MUNICIPAL COLLEGE OF TECHNOLOGY,
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June, 1925.

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CONTENTS

CHAPTER PAGE

PREFACE v

LIST OF SYMBOLS USED IN TEXT. ix

I. INTRODUCTION 1

II. GAS ENGINES AND THEIR CYCLES OF OPERATION . . 9

III. OIL ENGINES AND THEIR CYCLES OF OPERATION . . 34

IV. THE THERMODYNAMICAL RELATIONS OF A PERFECT GAS 59

V HEAT-ENGINE CYCLES OF OPERATION AND IDEAL EFFICIENCIES 78

VI. INDICATED MEAN EFFECTIVE PRESSURES, INDICATED HORSE-POWERS, AND CONSUMPTIONS OF IDEAL CYCLES ASSUMING CONSTANT SPECIFIC HEATS . . 101

VII. VARIABLE SPECIFIC HEATS AND THE ENERGY CURVES OF GASEOUS MIXTURES 111

VIII. ENTROPY, ENTROPY DIAGRAMS AND OTHER CHARTS . 125

IX. THE FORMATION OF MIXTURES REQUIRING EXTRANEOUS IGNITION 145

X. THE FORMATION AND FIRING OF MIXTURES IN WHICH AIR ONLY IS COMPRESSED, AND THE EXPANSION AND REMOVAL OF THE WORKING SUBSTANCE . . 166

XI. TEMPERATURE EFFECTS 178

XII. THE POSSIBILITIES OF DEVELOPMENT 198

NUMERICAL EXAMPLES 210

INDEX 229

LIST OF SYMBOLS USED THROUGHOUT THE BOOK

NOTE.—Where the meaning is obvious, capital letters are used to represent the larger units, and small letters the lesser quantities :

e.g. P = lb. per sq. ft. = $144p$, where p = lb. per sq. in.

A = area in sq. ft. *or* constant in variable specific-heat relation, $C_p = A + ST$.
a = area in sq. in.
B = constant in variable specific heat relation, $C_v = B + ST$.
B.Th.U. = British Thermal Unit of Heat.
C = specific heat in heat units, lb.-cal. per lb., or what is the same thing, gramme-calories per gramme.
C_p = specific heat at constant pressure.
C_v = specific heat at constant volume.
°C. = degrees Centigrade = $\frac{5}{9}$ (°F — 32°).
C.H.U. = Centigrade heat unit or lb.-calorie.
c = velocity in ft. per sec.
D = diameter in ft.
d = diameter in inches *or* the differential sign.
E = intrinsic or internal energy of a gas, *or* Young's modulus of elasticity.
e = base of the natural or hyperbolic logarithms.

$$e = 2{\cdot}7183. \quad \log_{10} x \times 2{\cdot}3026 = \log_e x$$
$$\log_e x \times 0{\cdot}4343 = \log_{10} x$$

°F = degrees Fahrenheit = $\frac{9}{5}$ °C. + 32°.
g = acceleration due to gravity in ft. per sec. per sec. ; taken as 32·20.
H = total heat available.
H.P. = horse-power.
h = total heat rejected.
I = constant pressure energy or "total heat."
J = Joule's mechanical equivalent of heat, taken as 778 ft.-lb. per B.Th.U. or 1400 ft.-lb. per C.H.U.

K = specific heat in work units = JC.
K_p = specific heat at constant pressure.
K_v = specific heat at constant volume.
k = thermal conductivity.
L = length in ft. l = length in inches.
M = number of impulses per minute.
m = molecular weight of a gas.
N = revolutions per minute.
n = index in the relation PV^n = a constant.
O = too much like the figure 0 to be used in written notes.
P = pressure in lb. per sq. ft. absolute.
p = pressure in lb. per sq. in. absolute.
Q = quantity of heat available per unit weight (or volume) of a gas, and therefore the calorific value of a gas when measured at S.T.P.
R = the gas constant in work units.
r = the compression ratio, or the ratio

$$\frac{\text{cylinder volume before compression}}{\text{cylinder volume after compression}} = \frac{V_2}{V_1} \text{ in figure below.}$$

S = the constant in the variable specific-heat relations (*see* A and B above).
S.T.P. = standard temperature and pressure.
T = the absolute temperature = $273 + {}^\circ C = 460 + {}^\circ F$.
t = temperature reckoned from 0°C or 0°F.

U = work done in ft.-lb. = $\frac{U}{J}$ in heat units.

V = volume of a gas in cu. ft.
v = specific volume of a gas. The volume occupied by unit weight, *i.e.* by one lb. at S.T.P.
W = weight of a gas in lbs.
w = specific weight of a gas. The weight of unit volume (1 cu. ft.) measured at S.T.P.
X = thickness *or* pressure ratio of expansion.
x = any unknown quantity.
Y = air to fuel ratio.
Z = time.

GREEK SYMBOLS

α (alpha) $= \frac{p_4}{p_3}$ in accompanying figure. Also coefficient of linear expansion.

β (beta) $= \frac{V_3}{V_2}$ in accompanying figure.

γ (gamma) = the ratio of specific heat at constant pressure to that at constant volume. $\gamma = \frac{C_p}{C_v}$

For variable sp. hts. $\gamma = \frac{A + ST}{B + ST}$.

δ (delta) = density of a gas (usually compared to air).

ϵ (epsilon) =

η (ēta) = efficiency.

θ (thēta) = angle.

κ (kappa) =

λ (lambda) = ratio $\frac{S}{B}$ in variable sp. hts. (*see* B).

μ (mu) =

ν (nu) =

π (pi) = the ratio of the circumference of a circle to its diameter. $\pi = 3{\cdot}1416$, $\log \pi = 0{\cdot}4972$.

ρ (rho) = the "cut-off" ratio $= \frac{V_5}{V_4}$ in figure below.

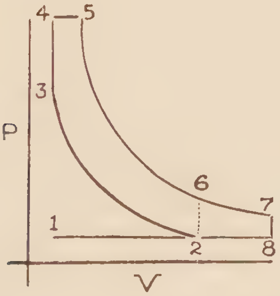
Σ (capital sigma) = the summation of finite quantities.

σ (sigma) = radiation coefficient.

ϕ (phi) = entropy.

ψ (psi) =

ω (omega) = angular velocity measured in radians per sec.



THE
ELEMENTS OF INTERNAL-COMBUSTION ENGINEERING

CHAPTER I

INTRODUCTION

The Internal-Combustion Engine and Some of its Difficulties.—The internal-combustion engine, as its somewhat cumbersome name implies, develops power from the combustion of a fuel within, or in close approximation to, the working cylinder itself. The fact sharply distinguishes this class of heat engine from the steam prime-mover, where the heating of the working substance takes place in a separate apparatus or boiler. In both cases, however, the working medium is in a gaseous state during the period when power is actually being developed, and a study of the laws of gases is equally necessary to both steam and internal-combustion engineering for a proper understanding of the cycles of events. The thermodynamics of steam are complicated enough, but the external-combustion engine has at least this advantage over its rival, that its working medium is, usually, confined to one substance. On the other hand, any kind of fuel which can form an explosive mixture with air is theoretically possible for use in an internal-combustion engine. The thermodynamics, to be of value, have to include the various ways in which the working mixture may be formed as well as the behaviour and state of the gas during the working stroke. This requires a knowledge of the physical and chemical properties of the fuel, the most efficient proportions in which it will combine with air to form an explosive mixture, the way in which it combines, the temperatures, pressures, and volumes which it attains during such combination, and the various cycles of operation which are possible in theory and in practice.

Of these, perhaps the most varied and certainly the most complicated is the proper formation of the working substance. A large amount of research work, which will be referred to in detail later, has been carried out with the express purpose of ascertaining the various ways in which suitable gases will unite on combustion. But the ground has not yet been covered and further information is still required as to the formation of the most suitable explosive mixture of air and combustible in the internal-combustion engine itself. The problem is of increasing importance now that a comparatively new type of oil engine is bidding fair to outstrip all rivals. In this engine, crude oil is mechanically injected direct into the combustion space by means of some form of oil-sprayer, but without the assistance of an air-blast such as is used in the Diesel engine proper. The difficulties of obtaining a thorough intermingling of the oil spray and the compressed air have only been partially overcome and, at the time of writing, the additional difficulty of extending the principle to the use of the heavier oil fuels in high-speed engines suitable for transport or aviation has barely been touched upon. It is only by a thorough study of the fundamental principles upon which such actions are constrained to work that a way out of such difficulties may be found.

Further Difficulties.—Another limiting factor, which, in the case of steam, has definitely drawn a line beyond which at present it has been found unwise to go, is the effect of high temperatures both on the materials used in construction and on the lubrication system of the machine. The present limit of such a temperature is about 400° C. or 750° F. and if it is exceeded *at a steady rate* for any length of time trouble is bound to occur. But temperatures in the working substance of an internal-combustion engine may reach 2000° C. (3600° F.) *momentarily*, and special precautions are essential to ensure that the mean temperature does not exceed some such limit as 400° C. The problem involves a study of heat transmission and distribution which is more complicated than in the case of steam.

It is possible to arrange for steam conditions which prevent a safe temperature from being steadily exceeded at the high-pressure end of the prime-mover. In fact, the limit is so seldom even reached that many steam-engine cylinders are jacketed to keep them warm, and so avoid premature condensation of the steam. An internal-combustion engine cylinder, on the other hand, has always to be jacketed or otherwise protected to keep it cool, and in some large sizes it has been found necessary to introduce complicated cooling arrangements for the piston and exhaust valves as well.

An additional difficulty arises from the fact that the working substance is ignited inside the engine itself, and if the temperature is even as high as 400° C. many mixtures will ignite before their proper time.

Temperature effects, however, both mechanical and thermodynamical, are, to a large extent, capable of analysis, and there exists a fair amount of research work to guide the designer of internal-combustion engines in the accuracy of his calculations.

It may perhaps be mentioned here that the question of this safe steady working temperature is one of the chief stumbling blocks in the evolution of a successful gas turbine.

The Human Factor in Design.—There is another influence at work, which is difficult to define, and which is seldom referred to in text-books, but which nevertheless frequently hampers the achievement of the best design, and that is what may perhaps be called a human or psychological factor.

When a new idea is being developed, there is usually a tendency to take some existing and successful idea and adapt it to the new conditions instead of sitting down and thinking out the significance of the new idea in all its aspects. This frequently leads to unnecessary complications and often tends to obscure the full advantages of the new method. The tendency is, of course, not confined to the development of internal-combustion engines, or even to engineering design in general, but seems to be ingrained in human nature.

Many instances may be given, such as the adaptation of the form of the road coach to railway carriages, or the early attempts to fly by means of flapping wings; but one or two examples from internal-combustion engineering may suffice to illustrate the point.

The early types of gas engines, such as those of Lenoir in 1860, and of Otto and Langen in 1866, used slide-valves for their gas and air inlets and for their exhausts. The Lenoir engine, as can be seen from its diagrammatic representation in Fig. 1, on p. 10, was avowedly copied from steam-engine practice, whilst the Otto and Langen (Fig. 3, on p. 13), though showing a commendable attempt to break away from traditional design in other respects, still retained the use of a slide valve. Even the Otto "silent" engine of 1876, which was the first internal-combustion engine to embody the all-important principle of compression in the working cylinder before firing, used a slide valve for the gas and air inlets (Fig. 5, p. 17), and the early engines of Sir Dugald Clerk (1880) followed suit.

But the slide valve as derived from steam-engine practice proved unsuitable for the different pressures and temperatures to be found in internal-combustion engines. Its application in the silent engine of 1876 is all the more interesting because Otto showed that he was familiar with the present type of mushroom valve by fitting one to the exhaust of the same engine.

Again, the first successful heavy-oil engine, that of Priestman, produced in 1888, was a direct application of the principle of the four-stroke gas engine in the stage which it had reached at that date. The oil was vaporised and mixed with air in a separate container placed inside the engine bed beneath the horizontal cylinder. The resulting gaseous mixture, with additional air, was drawn into the cylinder on the suction stroke and compressed before ignition by an electric spark in the usual way. It was left to Akroyd Stuart to introduce, two years later,[1] what is now characteristic of all heavy-oil engines, the injection and mixing of the oil with air in a chamber which forms part of the compression space of the cylinder.

Another typical example of this tendency in design can be found in the origin and development of the motor bicycle. Early machines, such as the Coventry Quadrant of 1906, were merely the application of a petrol engine to a strengthened bicycle frame, which still retained the ordinary chain-drive pedals for foot rests. Even now there is a tendency to build the engine into a bicycle frame instead of building a suitable two-wheeled carriage round the engine. An enlarged edition of the old bicycle seat is still retained—and the majority of back wheels are very inaccessible for tyre repairs.

---

[1] British Patent No. 7146 of 1890, in the names of H. A. Stuart and C. R. Binney.

The Scott two-stroke motor cycle and the machine known as the "Ner-a-car"[1] are perhaps the best attempts at present to break away from this restricted design.

This tendency to graft new ideas on to old, regardless of the suitability of the resulting arrangement, cannot always be avoided. Sometimes, in order to reduce the cost of manufacture, it is done with deliberate intention; but more forethought, combined possibly with a more thorough experimental stage and a more careful study of the fundamental principles under which the desired apparatus is constrained to work, cannot fail in the long run to make for better and more rapid progress.

Different Forms of Internal-Combustion Engines.—It is not intended in this book to describe in detail all the various types of internal-combustion engines which have been developed, but it is necessary to bear in mind that, starting with the Otto engine already referred to, many diverse methods have been put into practice with success to meet various conditions and requirements. There are perhaps two main reasons for this development. The internal-combustion engine gave promise, and so far has maintained that promise, of being the most efficient form of producing power from a given quantity of heat units in the fuel. Secondly, when the whole plant is taken into consideration, the internal-combustion engine can give the smallest ratio of weight per horse power developed of any practical form of prime-mover. This is more especially the case with the high-speed internal-combustion engine working on light spirit-oils, which has revolutionised road transport and made aviation possible. The main limiting factor, which enters chiefly into their use at sea, is the fact that at present the internal-combustion engine is essentially a reciprocator, with a limit to the power which can be developed from cylinders, so that in the case of very large powers the smallest weight-to-horse-power ratio no longer holds good.

The original engine of Otto in 1876 first embodied in a practical form three principles which have remained fundamental for the type ever since.[2] The first principle was the use of a compression stroke in the working cylinder before the gaseous mixture was fired; the second was the firing of this mixture at constant volume; whilst the third was the use of three pumping strokes to each working stroke. During these three strokes, the piston was driven by the energy which was stored in a heavy flywheel, and the strokes were used to exhaust the products of the previous combustion, to draw in a fresh charge, and to compress the charge before firing. An engine working on these principles is said to be working on a four-stroke cycle and on a constant-volume cycle, both of which are frequently called the Otto cycle. Occasionally, for brevity, such engines are loosely referred to as four-cycle engines.

---

[1] *Engineering*, vol. cxvi. (1923), p. 505.

[2] All these principles were first enunciated by A. Beau de Rochas in a French patent which he took out in 1862. This patent was abandoned in its second year and is not included in French patent journals. Beau de Rochas also published a brochure in Paris in the same year which is now very rare and practically unobtainable. He does not appear to have put his ideas to a practical issue and his work was probably unknown to Otto. For an abstract of the salient features of this brochure see "Les Nouveaux Moteurs à Gas," by G. Richard, Paris, 1892 ed. vol. i. p. 25. There is a copy of this book in the library of the Institution of Mechanical Engineers, London.

In the year 1880, Sir Dugald Clerk introduced a successful gas engine which gave a working stroke once every revolution.[1] This was done by the use of a second cylinder, or blower, which acted as a scavenging pump and which also introduced the fresh charge to be compressed in the working cylinder. This principle has also proved itself to be of great importance in internal-combustion engineering and is known as the two-stroke cycle or Clerk cycle.

Otto's name and principles are more closely associated with internal-combustion engines which work on gas, but the possible use of an " explosive-oil vapour " instead of a pure gas to form the explosive mixture is mentioned in his original patent of 1876. Subsequent experience has shown that, whilst gas engines and some types of oil engine have much in common, some modifications are advisable to get the best efficiency from each. But the fact remains that all gas engines, as foreseen by Otto, will work with vaporised oil to form the exploding medium, and need not differ in essentials, except as regards the arrangements for introducing the fuel into the cylinder. The Priestman oil engine, which has already been mentioned, illustrates this point. The converse, in some cases, is also true. This was typified during the great war of 1914-18, when, owing to the shortage of petrol, numbers of petrol-driven motor vehicles were converted to run on town gas. The gas was carried in balloon-like containers, which were filled from town supplies.

The Petrol Engine.—The petrol engine functions in the same way as the Otto or Clerk gas engines which have just been discussed. Arrangements have to be made to vaporise the oil, but as the spirit oils used volatilise at ordinary temperatures no heating device is required, and an apparatus known as a carburettor is all that is necessary to provide a gaseous mixture. The main difference between a petrol engine and its predecessors is the much higher rate at which the cycle of events takes place. This introduces factors in design which were absent in the earlier types.

G. Daimler is the real pioneer in this development of internal-combustion engines. His British Patent No. 4315 of 1885, although primarily intended to apply to gas, shows many of the features of a modern petrol engine, such as splash lubrication, partial crank-case compression, air cooling by means of a fan, half-compression and a ratchet crank handle for starting purposes. In the same year Daimler applied this engine to a bicycle (British Patent

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[1] British Patent No. 1089 of 1881. For his first attempts, see also his British Patents 3045 of 1878 and 2424 of 1879

No. 10786 of 1885) and in the next year to a launch (British Patent No. 14034 of 1886). Then in 1887 came the first petrol-driven motor carriage of which particulars are given by Sir Dugald Clerk.[1] By this time Daimler had invented the V-type twin cylinder engine, which was patented in Britain in 1889 (No. 10007). It was, however, not till the Locomotives on Highways Act of Britain was altered in 1896, that progress in this direction was feasible. Up to this date no mechanically-propelled vehicle could proceed along a public road in Britain unless a man carrying a red flag walked in front of it.[2]

The Diesel Engine.—The development of the internal-combustion engine had proceeded along the lines laid down by Otto, and to a lesser extent by Clerk, and the thermodynamic cycle upon which it worked was in both cases a constant-volume one; that is to say the working mixture of air and fuel was compressed in the cylinder and fired by means of a flame or an electric spark at the end of the compression stroke. The result was that the fuel burnt so rapidly that the maximum pressure (and temperature) was reached before the piston had time to move outward on the working stroke. Such a cycle is often referred to as an explosion cycle, and the next great step, that of Daimler to increase the speed from a maximum of 300 r.p.m. to 800 r.p.m. and over, still utilised a cycle of approximately the same nature.

In 1893 R. Diesel[3] proposed that air only should be compressed to such a high pressure that the heat of compression would in itself be sufficient to fire oil fuel when forced in under an air blast near the end of the compression stroke. Diesel's original ideas, which included a suggestion to work on the Carnot cycle, underwent considerable modification, but the first satisfactory Diesel engine was running by 1897 and the principle upon which this engine worked eventually gave the highest thermodynamical efficiency of any type of heat engine. One result of Diesel's method was to introduce the fuel gradually and burn it at more or less constant pressure, and the cycle is often called a slow-combustion one in contra-distinction to the explosive combustion of the Otto or Clerk cycles.

The "Semi-Diesel" Engine.—This term was somewhat loosely applied to oil engines which compressed their air only, but to a lower pressure than Diesel found necessary for spontaneous ignition, and which made up the deficiency by spraying their fuel mechanically into a hot bulb which formed part of the combustion space of the engine. In this way lower stresses were obtained than were inherent in the design of the pure Diesel, and many variations of the type were produced. The term "Semi-Diesel," whilst useful as a trade designation, was erroneous in its description of the type, as the cycle on which such engines were found to operate was the constant-volume cycle and not the special form of constant-pressure cycle characteristic of the Diesel.

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[1] D. Clerk, "The Gas and Oil Engine," 1902 edition, p. 460.

[2] Sir Alfred Yarrow is credited with being responsible for this law. He drove a steam carriage of his own invention in 1861, which unhorsed a mounted policeman when it was travelling at 25 miles per hour. See "Alfred Yarrow, his Life and Work," 1923, pp. 37 *et seq.*

[3] "The Rational Heat Motor." English edition, Spon. See footnote, chapter iii. p. 43.

The Dual-Combustion, or Mixed, Cycle.—More recently W. J. Walker[1] and others have drawn attention to a dual-combustion cycle in which the fuel is burnt partly at constant volume and partly at constant pressure. This cycle is the one followed in a number of modern oil engines which are variously termed: solid-injection, high-compression, cold-starting, sprayer, airless-injection, or mechanical-injection types. This dual combustion is obtained either by means of retarded injection or by the use of two or more separate oil-nozzles, and the engines, as some of their names suggest, depend upon the sprayer to atomise and pulverise the fuel without the use of an air-blast. The development of this type of engine, which bids fair to outstrip all its rivals, only took place during and since the great war of 1914–18, though experimental engines were constructed before that date.

The Still Engine.—In 1919 a paper was read in England by F. D. Acland[2] which made public, for the first time, a description of a new principle applied to internal-combustion engines, which had been developed during the war period by W. J. Still.

The range of temperature covered by the internal-combustion engine is much greater than is possible in the steam engine, but even so the lowest temperature, or heat rejected to the jackets and to the exhaust, is still comparatively high. The principle underlying the Still engine is the use of this heat to generate steam under pressure and apply it to the underside of the working piston. In other words, the Still engine attempts to combine the superior thermal efficiency of the internal-combustion engine cycle with the flexibility and mechanical advantages of the steam engine. Many forms of internal-combustion cycles have been tried by Still and his associates in the application of this principle, and it is interesting to note that the most recent developments to date (1924) are a marine type of engine which uses the solid-injection (D.C. two-stroke cycle) on the internal-combustion side, because the inventor considers this the simplest and best form.[3]

The Gas Turbine.—It is natural, by analogy with the steam turbine, that attempts should be made to develop a rotary form of internal-combustion engine. Many such attempts have been made in the past, and elaborate developments of an experimental nature are taking place both in this country and abroad. So far the practical results have been disappointing, chiefly on account of the temperature effects which have already been referred to, and it remains to be seen whether a successful and reliable gas turbine will eventually be evolved.

---

[1] Proc. Inst. Mech. Eng., Dec. 1920.

[2] Paper by F. D. Acland before the Royal Society of Arts on May 26, 1919. See also *The Engineer*, vol. cxxvii. (1919), p. 540, and vol. cxxxiii. (1922), p. 181.

[3] "The Type of Still Engine required for Marine Service with Special Reference to the Scott-Still Engine," by W. J. Still. Trans. N.E. Coast, Inst. of Eng. and Shipbuilders, vol. 40 (1924), p. 408.

There is, however, an application of the rotary principle which has proved itself to be useful in aviation. The power of any engine working from the atmosphere, will naturally fall off as the density of the atmosphere decreases. This fact sets a limit to the height to which aeroplanes can fly. Prof. Rateau,[1] in France, developed a small turbine which was driven by the exhaust gases of the petrol motor and which is therefore called an exhaust-gas turbine. This turbine, in turn, drives a small air compresssor, which makes up to the normal the pressure of an attenuated atmosphere. The method is known as "super-compression" and enables aeroplanes to fly higher than they otherwise could.

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[1] "The use of the Turbo-Compressor for attaining the Greatest Speeds in Aviation," by A. Rateau, Proc. Inst. Mech. Eng. (1922), p. 795.

CHAPTER II

GAS ENGINES AND THEIR CYCLES OF OPERATION

General.—Many attempts have been made to classify both gas engines and oil engines. Whilst the former have so far presented fewer difficulties in this respect than the latter, there still exists a difference of opinion as to the most convenient form such a classification should take. All modern gas engines function on the constant-volume cycle with compression of the working substance, but a broad distinction can be made which depends on whether the engine completes this cycle in four strokes or in two strokes; in other words, on whether in general it follows the Otto cycle or the Clerk cycle.

There is, however, much more diversity of type when the mechanical design of gas engines is considered. Some work with one side of the piston only and are therefore called single-acting. Others use both sides of the piston and are known as double-acting engines. In some, the cylinder or cylinders are arranged vertically, in others, horizontally. Some have cylinders placed one behind the other on one piston rod (tandem). Others have two pistons working in opposite directions in one cylinder (opposed piston type). One successful design works without a piston at all (Humphrey gas pump).

Constructional details of such variations in mechanical design are best studied in books specially devoted to this branch of the subject.[1]

In this chapter it is proposed to study the various cycles of operation which have been used in gas engines together with purely diagrammatic arrangements of the types concerned. In this connection it is well worth bearing in mind Sir Dugald Clerk's words, " As a rule of almost general application, specifications are untrustworthy as accurate descriptions of working machines." [2] To be of value, published drawings should be either detailed working drawings of machines in actual use, or pure diagrams to show the principles on which a machine works. In the latter case, which is adopted throughout this book, no reliance can be placed on the relative proportions or on the positions of the various parts, and frequently, for convenience, two or even three planes are shown in one elevation. The method has, however, the advantage of clearness when first studying the underlying principles of operation, and students are able more easily to make sketches for themselves.

---

[1] As for instance Güldner's " The Design and Construction of Internal-Combustion Engines," English translation by H. Diederichs, London, 1910. R. E. Mathot's " The Construction and Working of Internal-Combustion Engines," translated from the French by W. A. Tookey, 1910. H. Haeder's " A Handbook on the Gas Engine," translated from the German and edited by W. M. Huskisson 1911.

[2] D. Clerk, " The Gas and Oil Engine," 1902 edition, p. 118.

The Lenoir Gas Engine of 1860.—This engine was the first successful application of the principles of internal-combustion engineering to the development of power. It was modelled extensively on existing steam-engine practice and was very inefficient. It required about 100 cu. ft. of town gas per I.H.P. per hour. It was made in small sizes only, from half a horse-power to three horse-power, and had one horizontal cylinder with a "semi" double-acting piston. The piston-rod was connected through a cross-head to a crankshaft, which carried, on one side, a comparatively heavy fly-wheel.

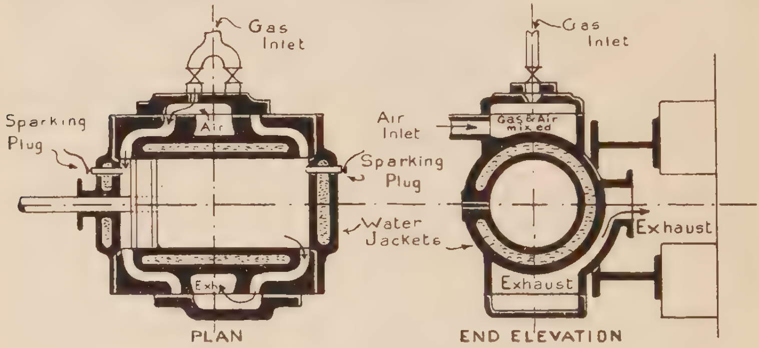

FIG. 1.—The Lenoir Gas Engine of 1860.

A diagram of a sectional plan and of an end elevation may be seen in Fig. 1. The details follow those given in Lenoir's British Patent No. 335 of 1860, which was taken out for him by J. H. Johnson. The first engines were made in Paris.

The gas came in at one side of the engine through a pipe which branched to two gas cocks, for feeding either side of the piston. When running normally, both these cocks were left full open, but they provided a means of controlling the gas supply by hand. The gas passed through a port hole in the top of a D-shaped slide valve, inside which it met and was mixed with air. The air was brought in through a filter by the air inlet on the top of the valve chest as shown in the end elevation. The gaseous mixture, which varied from about $\frac{6}{1}$ to $\frac{12}{1}$ in air to gas ratio, was fed through ports to each side of the piston in turn. In the meantime the opposite side of the piston was in communication throughout the whole stroke with the exhaust on the other side of the cylinder. This exhaust was also controlled by means of a D slide valve as shown. Both slide valves were driven by eccentrics from the main crankshaft as in a steam engine.

The cylinder was water jacketed at the sides and at both ends as far as possible. Ignition was by means of sparking plugs which were worked from a Ruhmkorff induction-coil, a Bunsen battery and a commutator rotating on the free end of the crankshaft.

It appears that trouble was experienced with both the water-cooling arrangements and with the electric ignition. Sir Dugald Clerk, who tested one of these engines, says that the piston often became red-hot because the small surface exposed to the water jacket by its periphery was not enough to dissipate all the heat absorbed by the two faces of the piston.[1] This occurred in spite of the fact that the piston speed was very low, only about 160 ft. per minute. Electric ignition generally was not at first found to work satisfactorily, even when the gaseous mixture was fired at atmospheric pressure. One of the chief improvements introduced into the Hugon engine [2] of 1865, which was otherwise very similar in design to the Lenoir engine and gave the same type of indicator card, was the replacement of the electric spark by flame ignition. Hugon also suggested the use of a water spray inside the cylinder to help in the cooling of the piston.

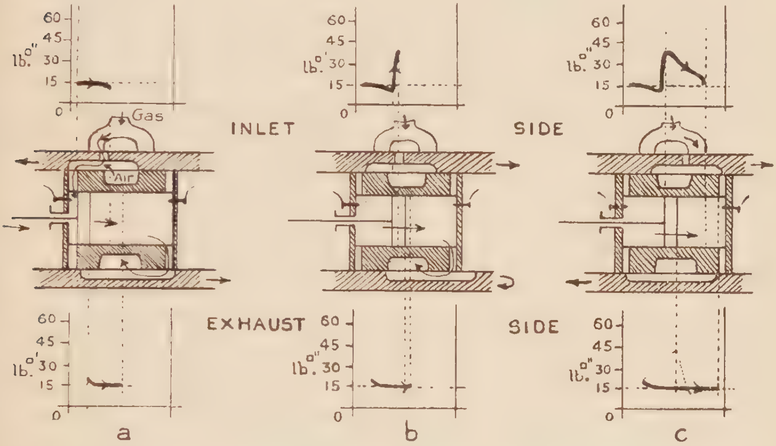

FIG. 2.—Two-stroke Cycle of Lenoir's "Semi" Double-acting Gas Engine.

Left of piston: (*a*) suction to half-stroke, (*b*) firing at half-stroke, (*c*) expansion to end of stroke.
Right of piston: (*a*) exhaust, (*b*) exhaust, (*c*) exhaust.

The Cycle of Operation of the Lenoir Gas Engine.—The type of indicator card, and therefore the way in which the Lenoir gas engine developed power, can be studied in Fig. 2. Consider a complete inlet stroke of the piston moving from left to right. At the commencement of this stroke the left side of the gas-pipe branch is opening to the inlet valve chamber. Gas is free to enter, mix with air, and fill the clearance space behind the left face of the piston. The piston moves to the right for approximately half its stroke by the energy stored in the flywheel, and in so doing draws in the gaseous mixture. At first this is at atmospheric pressure, but as the gas-port begins to close, the pressure inside the cylinder drops to one or two pounds below atmosphere. This action is shown in the diagram above, Fig. 2*b*. At this point the inlet valve closes, the ignition plug is timed to spark and the pressure inside the cylinder rises to its maximum (Fig. 2*b*, top diagram). The piston, however, is moving near its maximum velocity and explosion does not, therefore, take place quite at constant volume. After being fired, the gases expand and provide an impulse for the remainder of the piston's stroke (Fig. 2*c*, top diagram). At the end of the stroke the gases have expanded to within a few pounds of atmospheric pressure and the exhaust valve underneath opens on the left-hand side. The return stroke can best be seen from what happens on the right hand of the piston in the former stroke. This is shown (in Fig. 2) in the three diagrams underneath. The exhaust valve is open for the whole stroke and exhaust takes place at approximately atmospheric pressure throughout. The whole cycle is repeated on the return stroke from right to left. The indicator cards for the two sides are shown completed in dotted lines. It will thus be seen that, although the engine is double-acting, it only receives an impulse for approximately a half of each working stroke. It is therefore better described as "semi" double-acting.

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[1] D. Clerk, "The Gas and Oil Engines," 1902 edition, p. 122.

[2] P. Hugon, British Patent No. 986 of 1865.

The theoretical aspects of this card will be discussed later (p. 79).

The Otto and Langen Free-Piston Gas Engine of 1866.—Another engine which has both historical and theoretical interest is shown diagrammatically in Fig. 3. This represents the engine referred to in British Patent No. 434 of 1866, which was taken out by C. D. Abel on behalf of two Germans, E. Langen and N. A. Otto. The type is usually known as the Otto and Langen free-piston gas engine. The principle upon which it worked, however, was first invented by two Italians, E. Barsanti and F. Matteucci, whose British Patent No. 1655 of 1857 covered all the essentials of the type. They do not appear to have made a success of their ideas, whereas many Otto and Langen engines were made in sizes up to about 3 horse-power. The engine was very noisy and had many drawbacks, but it proved to be much more economical in gas consumption than any of its predecessors. A two horse-power engine tested by Sir Dugald Clerk[1] required only 25 cu. ft. of town gas per indicated horse-power per hour as compared with about 100 cu. ft. for the Lenoir engine of 1860.

The engine consisted of an inverted vertical cylinder standing on its base, open at the top and well surrounded with a jacket of cooling water. Across the top of this cylinder was mounted a free running shaft which carried a flywheel and a driving pulley. There was no crank, but the piston rod had rack teeth all down one side which drove a pinion mounted on the flywheel shaft. This pinion was arranged so that it would only engage when rotating in one direction. Its design was very similar to the free wheel of the modern safety bicycle. As the piston fell the pinion was in gear and rotated the flywheel, but when the piston returned on the up stroke the free wheel came into action and the flywheel continued to rotate in the same direction as before.

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[1] D. Clerk, "The Gas and Oil Engine," 1902 edition, p. 141.

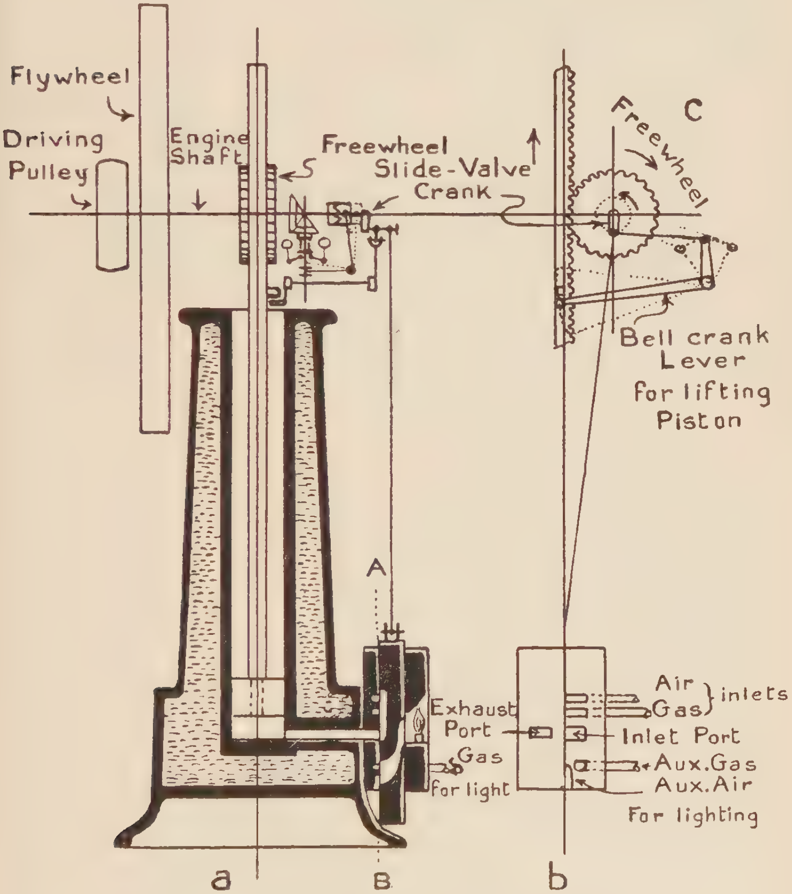

FIG. 3.—Otto and Langen Gas Engine of 1866.

A centrifugal governor, driven by bevel wheels from the engine shaft, connected a small eccentric into gear on to the engine shaft when the speed of the free-running flywheel dropped below a predetermined amount. This action is shown in the diagram by a claw coupling—though actually a pawl and ratchet gear was used for the purpose. The eccentric, when in motion, operated a single slide valve and also lifted the piston through the first portion of its upward stroke by means of a bell-crank lever which can be seen in the end elevation of Fig. 3*c*. The slide valve worked between two fixed faces, one of which is shown on the right-hand side of the figure (Fig. 3*b*). Air and gas were brought to the upper two slots in this valve face and transferred, at the proper time, through a D-shaped passage in the slide valve to the inlet port immediately below them. At the bottom of this same valve face were two further inlets for air and gas which were used for the ignition. The ignition port, which can be seen in the slide valve, opened at one end to this auxiliary gas and air supply, and at the other to a naked gas flame burning in a slot in the opposite, or outside, valve face. The consequence was that an " intermediate " flame was carried by this port as the valve moved upwards to the opening of the inlet port and fired the main gaseous mixture underneath the piston. This ignition port was so designed that the slot containing the gas flame was then shut off, so that explosion took place in an enclosed space. The exhaust port from the cylinder was placed alongside the inlet port, as can be seen in Fig. 3*b*, and was uncovered by a corresponding slot in the same slide valve, which is not shown in the diagram. An automatic valve was arranged at the exhaust exit to open against atmospheric pressure.

The action of the engine was as follows. When the engine was running slow enough to put the eccentric into gear, the bell-crank lever raised the piston by the energy stored in the flywheel for about one-twelfth of the stroke from its lowest position. During this period, a mixture of gas and air was entering the cylinder at atmospheric pressure and the "intermediate" flame was being kindled in the flame port. The slide valve moved up, fired the cylinder contents and the piston was shot up by the explosion to the top of its stroke, the free wheel being in action. The piston on its downward stroke gave an impulse to the flywheel and then drove out the products of combustion. At the bottom of the return stroke the exhaust valve closed and the piston rested until the falling speed of the engine set it again in action. The piston in the 2-brake h.p. engine tested by Clerk weighed about one cwt. and functioned at full load about 28 times per minute giving an average speed of 70 r.p.m.

The recoil was considerable and strong solid foundations were necessary. The engine was in effect a gun placed on end.

The Cycle of Operations of the Otto and Langen Engine.—This is shown in Fig. 4. For the sake of clearness the exhaust port is shown on the opposite side to the inlet, but in the actual engine both ports were controlled by the same slide valve.

In Fig. 4*a* the piston is raised mechanically by the action of the flywheel through about one-twelfth of its stroke, and the gaseous mixture is drawn into the cylinder at atmospheric pressure. The auxiliary gas and air supply are being ignited in the flame port.

Fig. 4*b* shows the gas and air inlets closed and the flame port, closed to the naked flame, igniting the cylinder contents. The pressure inside the cylinder follows approximately the curve shown on the P.V. diagram alongside. The mixture burns at constant volume, although there is reason to think that a certain amount of after burning occurs, and the pressure falls very rapidly to about 10 lb. per sq. in. below atmosphere or 5 lb. per sq. in. (abs.). As the effect of the impulse on the piston dies away, the combined effects of gravity and the difference in pressure on the two sides of the piston towards the end of its upward stroke, gradually bring the piston to rest against a rubber buffer. The piston rack was, of course, firmly supported in a guide to ensure a vertical motion with the rack in constant mesh.

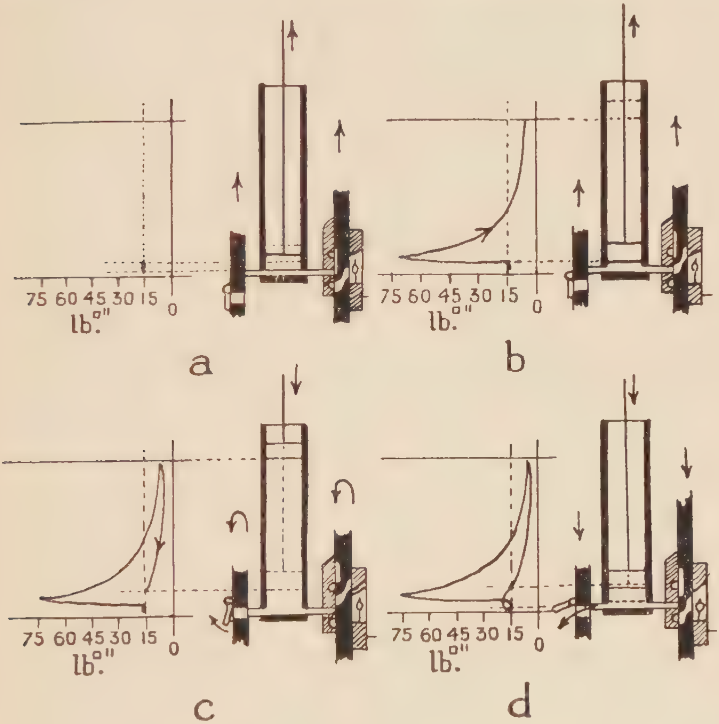

FIG. 4.—Cycle of Operations of Otto and Langen Engine of 1866.

(*a*) Inlet. (*b*) Ignition and expansion.
(*c*) Working portion of stroke. (*d*) Exhaust.

Fig. 4*c* shows the working portion of the return stroke. The flywheel is driven by the falling weight of the piston and the exhaust valve opens when the pressure inside exceeds that of atmosphere. A small automatic valve is fitted to the exhaust exit to ensure this. It is shown diagrammatically as a flap valve. In Fig. 4*d* the remaining portion of the downward stroke is shown. The piston drives out the burnt gases through the exhaust valve until the latter is closed by the action of the slide. This brings the piston to rest at the bottom of its stroke, and the cycle is repeated.

The theoretical aspects of this card are discussed on p. 82.

The Brayton Gas Engine and the Simon Steam-Gas Engine.—The principle of the Brayton engine was apparently first expounded in British Patent No. 432 of 1872, which was taken out in the name of W. R. Lake on behalf of G. B. Brayton. It introduced, for the first time, the idea of compression before ignition, though in this type of engine the compression took place in a separate pumping cylinder and not in the working cylinder itself. The pumping cylinder was driven by the energy stored in a flywheel and delivered a mixture of air and gas to an intermediate receiver at a pressure of about 75 lb. per sq. in. (abs.).

From the receiver, the compressed gas and air passed into the working cylinder for the first portion of each outward stroke. This cylinder was single-acting and received one impulse every revolution. A naked gas flame was arranged to burn in front of a wire gauze at the point where the gaseous mixture was introduced, and, as a consequence, this mixture burnt at approximately constant pressure during the whole period of admission. The wire gauze worked on the principle of the Davey safety-lamp for mines, and prevented the flame from striking back into the reservoir during the firing portion of the stroke. The arrangement was not very successful for firing gas, and Brayton, who developed his engine in America in 1873, converted it to work on petroleum, and so made one of the pioneer oil engines. The indicator card was very similar in appearance to a steam-engine diagram. There was no explosion such as occurs when the mixture burns at constant volume, and the engine utilised the principle of slow-combustion which was afterwards applied, in a different manner, by Diesel.

The Brayton engine was introduced in this country about the year 1878 by L. and R. Simon,[1] who modified it by passing the exhaust products through a spiral coil which was surrounded by water in a small closed vessel. This water got so hot that steam was generated, and Simon utilised this steam by introducing it into the working cylinder along with his gaseous mixture. The idea was to assist in the impulse and, at the same time, to lubricate the piston and keep it cool. It was, in fact, an early type of "spray-injection," though, in this case, the spray was in the form of steam. The method did not prove feasible, but it is interesting as showing an early attempt to utilise the heat which would otherwise have been lost in the exhaust. The modern Still engine avoids the drawbacks of Simon's gas engine by keeping the steam and gaseous mixture on separate sides of the piston.

The Otto Silent Gas Engine of 1876.—The first internal-combustion engine to embody the all-important principle of compression in the working cylinder before ignition, is shown in British Patent

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[1] British Patent No. 433 of 1878. L. and R. Simon.

No. 2081 of 1876. This patent was taken out in the name of C. D. Abel on behalf of N. A. Otto, and the engine illustrated there is shown diagrammatically in Fig. 5. The engine was called Silent by its inventor to distinguish it from the Otto and Langen engine of Fig. 3, which was just the reverse. It is the father of all subsequent gas engines in that it embodied principles which are still in use.

The engine consisted of an overhung horizontal cylinder, mounted on to a bed plate, closed at the rear, or breech, end but open at the crankshaft end. The cylinder was surrounded by a jacket of cooling water and contained a piston either of the trunk type or occasionally of the box type. The crank pin is shown in the patent drawings directly connected to a gudgeon pin in the piston and there is no piston rod, but many of the earlier engines still retained the crosshead of the steam engine. The gaseous mixture was introduced and fired by means of a slide valve moving across the breech end at right angles to the motion of the piston. This slide valve was driven through a small connecting rod and crank on the end of a "half-motion" shaft, which also carried a cam to operate the mushroom exhaust valve through a rocker lever. The half-motion shaft itself was driven by means of 2 to 1 reduction bevel gears from the crankshaft, on the opposite side of which the flywheel was fixed. The crankshaft had two webs, one each side of the crank pin, and was mounted on two bearings. Outside the heavy flywheel, a driving pulley was fixed in a similar way to the Otto and Langen engine of Fig. 3.

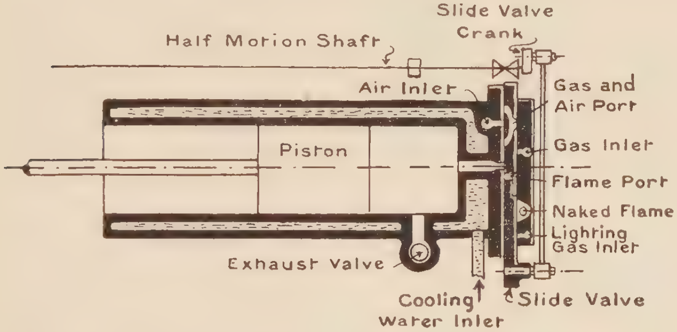

FIG. 5.—The Otto "Silent" Gas Engine of 1876 (in plan).

The slide valve worked between two fixed valve faces. The one nearer the engine had a port on one side for inlet air and a central communication with the combustion chamber. The outside fixed face carried three openings—a port for main gas inlet, a chamber containing a naked gas flame, and another, and smaller, port for both gas and air to be used for ignition purposes. The slide valve itself had a D-shaped passage, marked "gas and air port," which spanned the air inlet and the entry to the combustion chamber at the same time as an opening in the back of the D registered with the working gas inlet. Another opening straight through the slide valve is marked "flame port." It had a short passage on the outer side of the slide valve which linked up, at the appropriate moment, the lighting-gas inlet with the naked flame. The function of the flame port was to fill with gas from the lighting gas, ignite, and carry the small charge of burning gas to the entry of the combustion chamber after the latter had been filled with a suitable mixture of compressed air and gas.

The exhaust was quite separate. It was placed at one side of the compression space and consisted of a conical-seated mushroom type of valve which was lifted and closed against a spring by means of a rocking arm, passing underneath the cylinder and manipulated by a cam mounted on the half-motion shaft.

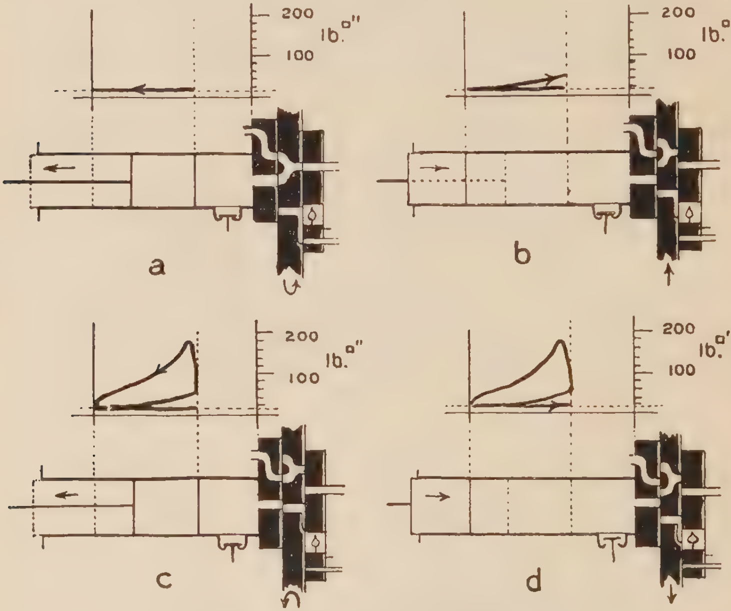

FIG. 6.—Cycle of Operations of Otto's "Silent" Gas Engine.

(*a*) Suction stroke. (*b*) Compression stroke.
(*c*) Working stroke. (*d*) Exhaust stroke.

The engine was governed on what is now termed the "hit-and-miss" principle. If a certain predetermined speed is exceeded, the governor causes the gas supply to be cut off and air only is drawn into the cylinder for the succeeding working stroke. This causes the engine to lose power and as the speed falls the gas supply is automatically restored.

The engine was single-acting and required three pumping strokes for each working stroke. The original engines were made in sizes up to about 12 B.H.P. when running at about 160 r.p.m. They had a compression ratio of $2\frac{2}{3}$ to 1 and an air to fuel ratio of about 7 to 1. Their consumption in gas, including that required for ignition, was about 22 cu. ft. of British town gas per I.H.P. per hour. This was as good as, if not better than, anything that had been done before.

Cycle of Operations of the Otto "Silent" Gas Engine.—The engine is single-acting, that is to say the working fluid acts on one side of the piston only, the other side being open to the atmosphere. There are four strokes in the complete cycle, represented by the four diagrams in Fig. 6.

In the first stroke shown (Fig. 6*a*), the piston travels from its innermost position to its outermost point by the energy previously stored up in the flywheel. The exhaust valve is closed and the slide valve is timed to move near the end of its stroke so that the air and gas inlets are both in communication with the working cylinder, and the mixture is drawn in by suction at approximately atmospheric pressure. At the same time, the lighting gas is placed in communication with the naked flame and ignited in the flame port of the slide valve.

In the second stroke (Fig. 6*b*) the exhaust remains closed and the slide-valve has moved sufficiently to cover all the inlet ports. As a consequence, the working substance has no means of egress and is compressed by the returning piston, which is still driven by the flywheel.

At the end of this compression stroke the flame port reaches the inlet passage to the working cylinder, and as it contains burning gas, the whole contents are ignited and explode at approximately the constant volume of the compression space. This is followed by the working stroke of the cycle (Fig. 6*c*), the products of combustion, after reaching their maximum pressure and temperature, expanding and forcing the piston outwards to do work on the crankshaft.

Near the end of the working stroke, the exhaust valve opens for the first time during the cycle and remains open for the whole of the second return stroke, whilst the slide valve is returning to its original position to admit a further charge of air and gas at the beginning of the next cycle. The piston moves from right to left and drives out the burnt gases at approximately atmospheric pressure. This fourth stroke is consequently called the exhaust stroke of the cycle (Fig. 6*d*).

It will be noted that, in order to achieve the cycle of events just described, the slide valve must only make two strokes whilst the working piston is making four. The exhaust valve also only functions once in four strokes, that is in two complete revolutions of the crankshaft. Hence the arrangement of a lay shaft which revolves at half the speed of the engine proper, an arrangement which is still very common in internal-combustion engines of all kinds.

The Clerk Two-Stroke Gas Engine of 1880.—Sir Dugald Clerk, who was one of the pioneers of internal-combustion engineering in this country, and who is the greatest living authority on the subject at the present time (1924), was the originator of the first successful gas engine which had a working impulse once every revolution from a single cylinder. The engine formed the subject of British Patent No. 1089 of 1881, but was first successfully made towards the latter end of 1880. It is shown diagrammatically in Fig. 7. There were two cylinders, one for developing the power and one for preparing the fresh charge and pumping it into the working cylinder at the proper time. Each of the cylinders was fitted with a trunk type of piston, and was therefore single-acting, and both pistons were connected to the single crankshaft. The piston in the pumping cylinder, or displacer cylinder as it was first called by the inventor, worked about 90° in advance of the main piston. Both cylinders were horizontal and lay side-by-side, whilst the working cylinder was well jacketed with cooling water. The pumping cylinder did not require jacketing, as it only dealt with comparatively cool gases.

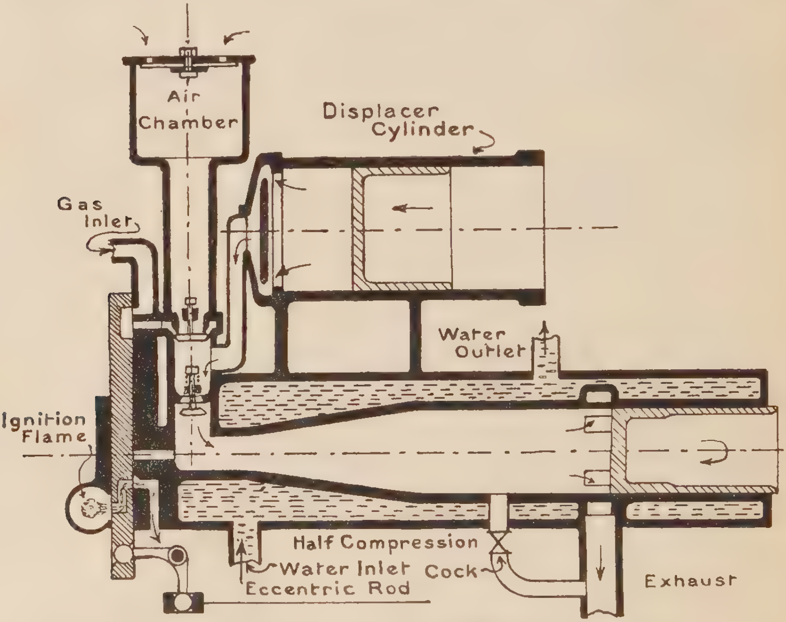

FIG. 7.—Clerk's Two-Stroke Gas Engine of 1880 (composite plan).

Across the breech end of the main cylinder a slide valve was placed, which was driven from the crankshaft through an eccentric and a bell-crank lever. This slide valve carried a flame port for the ignition and a slot to cut off the incoming gas when required. The fuel gas passed first through this slot and then to an annular space in the air duct, which formed the seating for a mushroom valve, and which had small openings drilled in it, uncovered by the valve. The air came into the air chamber through a rubber foot-valve and then intermingled with the incoming gas as it passed through this first mushroom valve. The suction was obtained from the pump piston which drew the mixture into the displacer cylinder on its outward stroke. The slide-valve slot was so arranged that the gas supply was cut off when the pump piston had reached approximately half-stroke, with the result that the remainder of the outward stroke drew air only into the displacer cylinder. On the return stroke, the contents of the displacer cylinder were discharged through the lower mushroom valve into the working cylinder.

In the actual engine, the air chamber was underneath and the air duct was consequently vertical. Both the mushroom valves were automatic and dropped back on to their seatings against gravity. A light spring, however, was fitted to the valve in the combustion head to hold it against a possible back pressure of some 5 lb. per sq. in., which was reached by the displacer piston at the very end of its inward stroke.

The compression space of the working cylinder was made conical, and a small port hole at the extreme inner end communicated with the flame port in the slide valve at the proper moment. There were no exhaust valves, but the piston itself uncovered a number of slots at the end of its outward stroke. These slots, or exhaust ports, opened into an annular passage surrounding the cylinder liner, which communicated underneath to the exhaust pipe. A bye-pass was arranged from the middle of the cylinder to this exhaust pipe to allow for half-compression when starting the engine.

The engine ran considerably faster than had previously been the case, the smaller sizes having a speed of 200 r.p.m., and consequently the old type of flame ignition device had to be re-designed to function with sufficient rapidity. This was done by providing a bye-pass to the atmosphere which kept the flame continuously burning, as shown in the figure. A small gas pipe from the main gas supply, which is not shown in the diagram, fed the flame port. As the slide moved across, the ignition flame and then the bye-pass to the air, were cut off in succession, and in this way sufficient burning gas was carried in the flame port to ignite the cylinder contents when it reached the combustion port.

This engine rapidly attained an economy equal to the best gas-engine practice of its day, and had a consumption of about 20 cu. ft. of British town gas per I.H.P. per hour. The ratio of compression of the early engines was 3 to 1, and the average proportion of air to gas was about 10 to 1.

It has proved itself the forerunner of an important branch of internal-combustion engineering, and although the difficulties are greater than would appear at first sight, there are many advocates and examples of two-stroke internal-combustion engines in existence at the present day.

Cycle of Operations of the Clerk Gas Engine.—The cycle was complete in two strokes of the engine and there was consequently one working stroke for each revolution. Fig. 8 (*a*) shows the working portion of the outward stroke. The flame port has reached the combustion chamber and fired the contents, which burn at approximately constant volume and then expand to about 30 lb. per sq. in. above atmosphere.

The displacer piston, having an advance of 90°, has reached about mid-stroke when ignition occurs in the working cylinder. The gas inlet has just been cut off and the piston completes its outer stroke (and a small portion of its return stroke), drawing in air only.

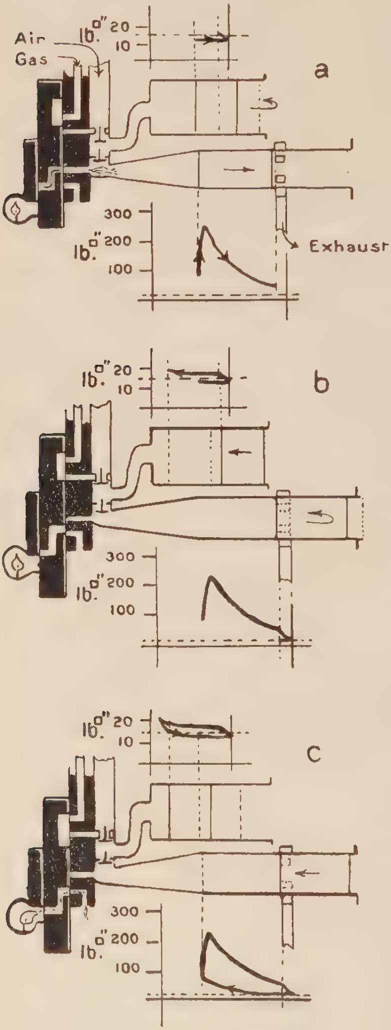

FIG. 8.—Cycle of Operations of Clerk Engine of 1880.

(*a*) Working portion of the cycle.
(*b*) Exhaust of the cycle.
(*c*) Compression of the cycle.

Fig. 8 (*b*) shows the exhaust portion of the cycle. The main piston uncovers the exhaust ports and the pressure drops to atmosphere. The displacer piston, having already started on its return stroke, continues throughout the exhaust to move inwards and drive the first portion of its contents into the main cylinder. In this way the products of combustion in the main cylinder are displaced, first by fresh air and then by a gaseous mixture for the next stroke. The volumes of the two cylinders are carefully proportioned to balance these two effects as nearly as possible, the bulk of the fresh air being used to drive out the burnt gases whilst the bulk of the incoming charge remains in the cylinder.

Fig. 8 (*c*) shows the compression portion of the cycle. This commences as soon as the exhaust ports are covered by the returning piston. The slide valve has closed the combustion port, but opened the gas-inlet port, so that the displacer piston, which is moving outwards for the major portion of the compression stroke, is drawing in air and gas for the succeeding charge.

In this engine, of course, the slide valve makes as many strokes as the piston and there is no half-motion shaft.

Atkinson "Cycle" Gas Engine of 1886.—This engine, although no longer made, is of considerable interest to students of internal-combustion engineering in that it successfully attempted to introduce a cycle of operations which can be shown in theory to be more efficient than either the Otto or the Clerk cycles (see p. 90). It was patented by J. Atkinson in Britain, in 1886 (No. 3522), and in it the inventor devised a link motion which had the effect of giving four unequal strokes to the piston in the cycle, instead of four equal strokes as in most other four-stroke gas engines. The way this was achieved can be seen from a study of Fig. 9, which is a diagram of the engine as published in *Engineering*.[1] Consider first the portions of linkage marked with the figure 1. The trunk piston has a connecting rod which is not directly coupled to the crank pin, but is connected to a rocking lever which is in effect an elongated triangle as shown, with a pin joint at each angle. The triangle is nearly isosceles in shape with two long sides and a very short base. The connecting rod joins one base angle and a toggle lever the other. This toggle lever is pivoted as shown and swings backwards and forwards through about a right angle. The apex of the rocking lever triangle is connected to the crank-pin proper, which rotates the crankshaft and flywheel in the same direction as the hands of a clock.

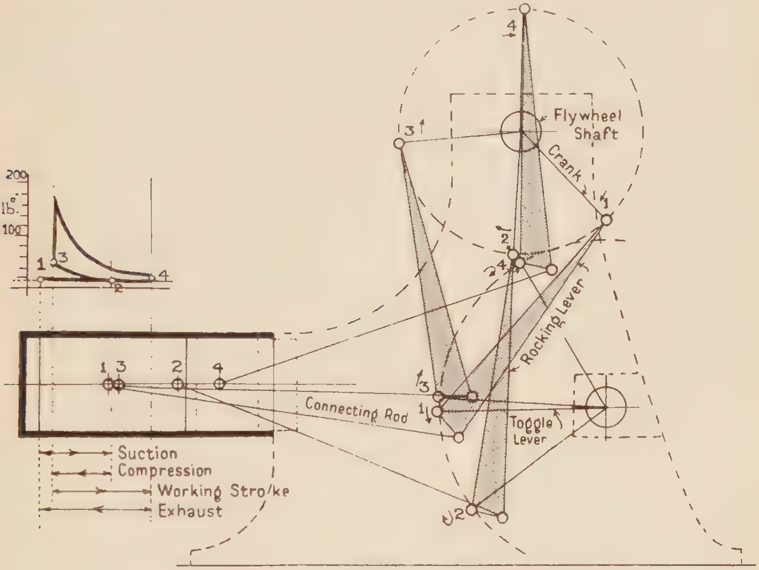

FIG. 9.—Atkinson "Cycle" Gas Engine of 1886.

The position 1 shows the piston in its innermost position at the commencement of the suction stroke. The rotation of the flywheel swings the apex of the rocking lever to position 2, with the result that the toggle is forced downwards and the piston moves out for the suction stroke.

As the flywheel continues rotating to position 3, the rocking lever moves upwards along the crank-pin path, and the piston makes an inward stroke of compression. Owing to the fact that the connecting-rod pin is eccentric to the pin of the toggle and that the rocking lever has now turned to the left about this toggle pin, the piston does not travel back quite so far on this compression stroke as it did for position 1.

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[1] *Engineering*, vol. xlvi. (1888), p. 627.

The motion between position 3 and position 4 is the working portion of the stroke, and it will be noted that it occupies about 100° of the crank motion. The piston travels much farther out on this stroke, as can be seen by the dotted lines projecting from the cylinder liner.

The return from position 4 to position 1 is the longest stroke of all, and represents the exhaust portion of the cycle. Here again, of course, the piston is driven by the energy stored in the flywheel.

The resultant indicator card is shown above the cylinder and the numbers correspond throughout.

It should be noted that although this engine is of the four-stroke-cycle type, the flywheel only makes one revolution per cycle.

In order not to complicate the diagram, the valve-gear as well as the ignition has been omitted from Fig. 9. There was no slide valve and the whole gear was of much simpler pattern than anything that had been made before. It may be studied in detail in the *Engineering* article already referred to.

The Atkinson engine came into prominence as the result of winning a trial which was carried out by the Society of Arts at South Kensington, in September, 1888.[1] In this trial an engine, when running at 131·1 r.p.m., developed 11·15 I.H.P., and had a gas consumption of 19·22 cu. ft. per I.H.P. per hour, using gas with a cal. val. of 633 B.Th.U. per cu. ft. The engine cylinder was 7½ in. in diameter, and the lengths of the four strokes were as follows: suction 6·33 in., compression 5·03 in., working stroke 11·3 in., and exhaust 12·43 in. The clearance volume (up to the point 1 on the card of Fig. 9) was 10 per cent. of the maximum stroke.

A considerable number of Atkinson Cycle engines, up to about 30 B H.P., were made in the ensuing five years; but it was found in the end that the number of joints, five as against two, led to mechanical difficulties in maintenance. The improvements made in that time with the Otto type of engine caused the Atkinson engine to be abandoned.

A Modern Gas Engine.—Modern gas engines are, for the most part, direct descendants of the Otto "Silent" engine of 1876. They have a single horizontal cylinder, are single-acting and work on the four-stroke cycle of Otto. The ratio of compression is now increased to about 5·5 to 1, and the maximum pressure to between 300 and 400 lb. per sq. in., or even higher. The slide valve is done away with altogether, and flame ignition has been replaced by a magneto-operated electric sparking-plug. The consumption on good quality British town gas is about 15 cu. ft. per I.H.P., and as much as 300 horse-power may be developed in one cylinder, though engines are not usually built for more than 125 horse-power per cylinder.

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[1] Report of the Judges, Society of Arts' Trials of Motors for Electric Lighting, 1888–9. There is a good summary of these trials in Robinson's "Gas and Petroleum Engines," 1890 edition, p. 528.

A diagram of a side elevation of a modern gas engine cylinder, which does not represent any particular make, is shown in Fig. 10. The cylinder and breech end are both well jacketed with cooling water. The gas and air inlets are separately controlled by butterfly-wing valves which may be actuated by the governor or by hand. In this way better control is kept over the air-to-fuel ratio, and the method is known as quantity governing. The gas and the air enter a valve cage through a lantern with a number of slots in it, and the gas valve is spring seated on the valve spindle of the inlet valve proper. This has the double effect of ensuring that both valves bed properly, regardless of possible wear on the seating, and it also enables the gas valve to open later and close earlier than the main inlet valve, should it be desirable.

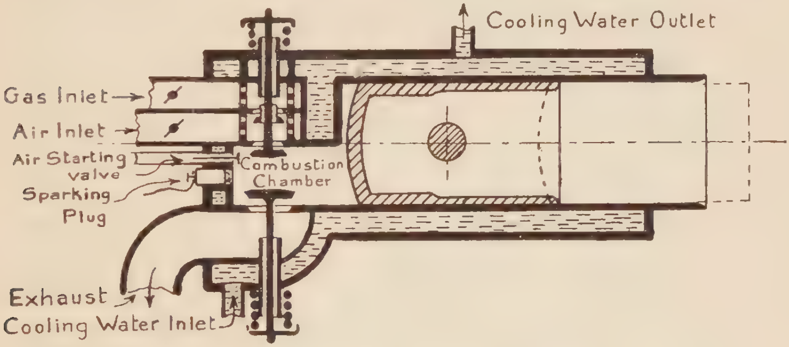


FIG. 10.—Modern Gas-Engine Cylinder.

Arrangements are frequently made to withdraw the whole of the inlet valve and its cage upwards, so that repairs can be effected with comparative ease and the valves conveniently ground in on their seatings. The exhaust valve in the larger sizes also has a separate seating, but access for grinding may be obtained from above when the inlet-valve cage has been withdrawn. The exhaust valve is placed underneath and both inlet and exhaust valves are actuated against springs by rocking levers which work from cams on a half-motion shaft driven by gearing from the crank-shaft of the engine.

Such engines are not self-starting. They are usually started by compressed air from an external source of supply, though the smaller ones can be started by hand. The small air-starting valve may be seen between the air inlet and the sparking-plug pocket.

The cooling water is usually introduced at the lowest point and drawn off at the top. The general symmetry of the design should be noted and the lack of end-pockets in the water jacket. This is an important point, as the cooling water supply is frequently of a dirty nature and any space where solid deposits could form might lead to trouble through burning or cracking of the metal walls.

The Cycle of Operations of the Modern Gas Engine.—This cycle, which is similar to that of the Otto Silent engine of Fig. 6, is shown in detail in Fig. 11. Although it is customary to label the four strokes of this cycle as suction, compression, ignition and expansion, and exhaust, the duration of each of these four periods is not quite equal. It may be varied at will by the setting or timing of the valves, and is not the same for all gas engines.

The inlet valve proper usually opens before the piston has reached its inner dead centre, as is shown in Fig. 11 (*a*), position 1, though the gas valve may be retarded to open at 1*a*. The suction stroke, Fig. 11 (*a*), takes place throughout the outward motion of the piston at an average pressure of ½ to 1 lb. per sq. in. below atmosphere. The gas valve should close near the end of this outward stroke (2*a*), but the inlet valve proper may close later (2).

Compression takes place on the return stroke, Fig. 11 (*b*), from position 2 to about position 3. The pressure reached at the end of compression will depend upon the ratio of the total volume (and pressure) of the gaseous contents when the piston is in position 2, to the corresponding total volume when the piston has reached position 3. In the illustration, this ratio has been made 5·5 to 1, and the corresponding compression pressure is shown at 140 lb. per sq. in. (abs.).

Ignition takes place at position 3, Fig. 11 (*c*), just before the piston has reached its inner dead-centre, and the gases explode at approximately constant volume to reach their maximum pressure almost immediately. What this pressure will be depends upon a variety of conditions which will be explained later. As the piston moves outwards again, the gases expand and do work until position 4 is reached.

The exhaust valve opens at position 4, Fig. 11 (*d*), and remains open until after the air inlet has opened again at position 1. The pressure drops to that of the atmosphere, and the incoming air at the end of the stroke tends to remove the residue of burnt gases from the combustion chamber itself. The exhaust valve closes at about position 5, or just before the gas inlet opens at 1 (*a*).

It must be understood that this description, like a good many others in a book of this nature, is merely typical of what happens, and should not be taken as applicable to every case.

Some Developments of the Modern Gas Engine.—The gas engine, for the most part, has remained in the form just described in the previous paragraph. It functions on town gas or on power gas made specially for it in gas producers. The discovery, about the year 1894, by B. H. Thwaite and others, of the possibility of using the gases which emerge from blast-furnaces, coke-ovens, or other forms of furnaces, has led to the design of very large-sized gas engines, which have taken various forms.

The main lines of development are shown diagrammatically in Fig. 12. They are usually known by the names of their first makers,

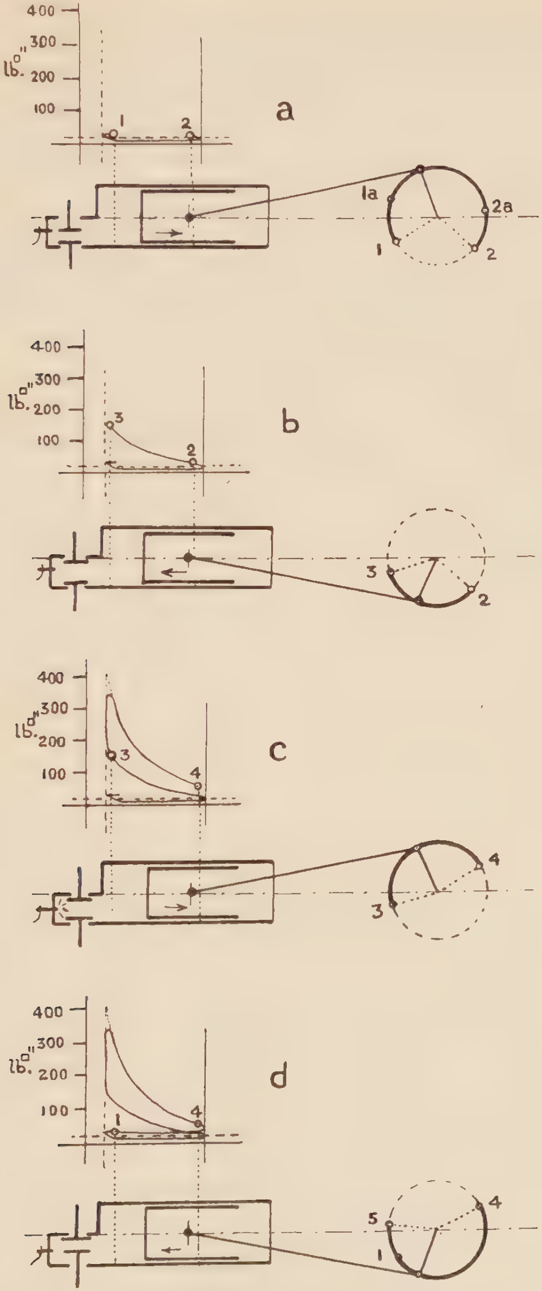
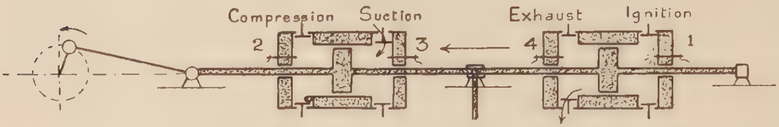


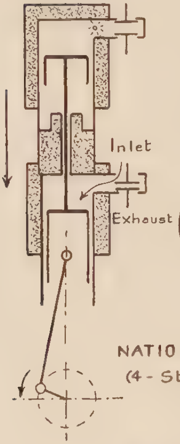
FIG. 11.—Cycle of Operations of Modern Four-Stroke Gas Engine.
(*a*) Suction. (*b*) Compression. (*c*) Ignition and expansion. (*d*) Exhaust.

though there are now several variations in different countries. It will be noted that two of the types work on the Otto four-stroke cycle, and that the other two are applications of the Clerk two-stroke cycle.

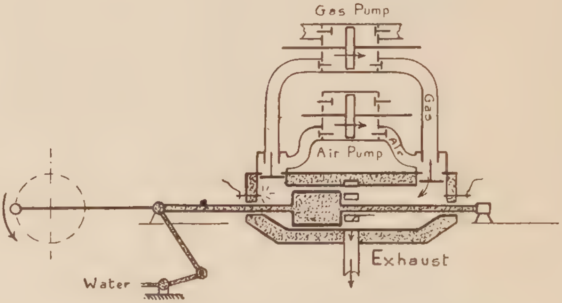
Of the two four-stroke engines the *Nuremberg* is double-acting, whilst the *National* is single-acting. By arranging two equal powered cylinders in tandem in the former engine, one working impulse for each stroke of the engine is obtained, the firing taking place as numbered.



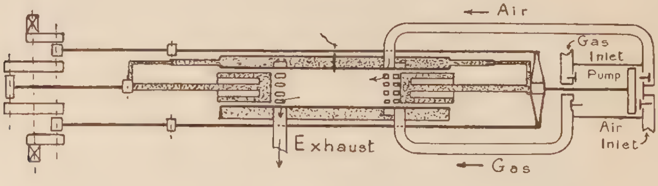
NUREMBERG (4-Stroke)



NATIONAL (4-Stroke)



KOERTING (2-Stroke)



OECHELHAUSER (2-Stroke)

FIG. 12.—Gas Engine Development.

The same result is achieved in the National single-acting engine, which is vertical, by placing two pairs of cylinders side by side on one crankshaft. The arrangement in the latter case may be extended to three or more pairs on the same crankshaft to obtain multiples of the original power.

The *Koerting* two-stroke engine has one cylinder, which is double-acting, and it follows that in this type there will also be one working impulse for each stroke of the engine. The appearance of this engine is similar to the uniflow steam engine, in that there are no exhaust valves, and the products of combustion leave the cylinder through central exhaust ports which are uncovered by the box-shaped piston. This represents a development of the method used by Sir Dugald Clerk in his engine (Fig. 2). The *Oechelhauser* two-stroke engine is interesting as being the first internal-combustion engine to use the principle of two single-acting pistons moving in opposite directions in a plain cylinder liner which is open at each end. The opposed-piston engine, as it is called, has found considerable favour in recent years in marine oil-engine design, and the Oechelhauser gas engine may be considered as the forerunner of these types.

The piston nearest the crankshaft is connected through a crosshead to the central crank pin. The other piston has a T-piece at the end of its main piston rod, to either end of which auxiliary piston rods are fixed. These rods are connected through side crossheads to two crank pins set at 180° to the central crank pin. The result is to get an engine with balanced reciprocating parts. It is, however, single-acting and therefore gives only one working stroke per revolution.

In both the Koerting and the Oechelhauser the action of the valves supplying the fuel is similar. Being two-stroke engines, some form of pump is necessary for scavenging the products of combustion from the cylinder and for introducing the fresh charge.

The pump valves are purely diagrammatic in Fig. 12. In practice they take various forms and are not necessarily of the mushroom type as shown. All these valves are mechanically controlled to open and shut at the correct time, and in both cases pure air enters the cylinder first to provide a scavenge. The Koerting has a mushroom type of inlet-valve proper, but the Oechelhauser has no valves in the cylinder, the air and gas inlets being separate rings of port holes, which are uncovered by one of the pistons towards the end of its outer stroke.

In all four types, ignition is by means of an electric spark, though frequently more than one sparking plug is supplied for each cylinder. The water cooling arrangements are indicated by the dotted areas. All the engines are jacketed as much as possible, and the National is the only one without water-cooled pistons as well. This prevents it being made with such large cylinders as are possible in the other cases. To get the water to and from the pistons, the piston rods have two water passages throughout their lengths, and the water is introduced and withdrawn through a rocking pipe connection to a crosshead or a slipper, which can best be seen in the Koerting engine. The Oechelhauser uses telescoping water tubes which are connected to the water jacket; these are shown above each piston rod.

The Humphrey Gas Pump.—There is one other successful application of gas to internal-combustion engineering which should be noted, and that is the gas pump which is the invention of H. A. Humphrey.

A diagram of the essential parts of the cylinder of the large pumps which are installed at the Metropolitan water-works at Chingford, is shown in Fig. 13. The lower end of this cylinder is immersed in water and bends round to a long play pipe with a riser at the far end. There is no piston, the surface of the water taking its place, and the pumping action is obtained by the explosion of a mixture of gas and air in the combustion chamber, or closed end of this cylinder upon this surface, which forces the whole column of water downwards and up to a greater height at the far end. A delivery pipe at this far end draws off some of the water and the remainder comes to rest and swings back to correspond to the second stroke of the cycle. The pump is designed to work on a four-stroke cycle, and this is obtained by allowing the water to oscillate freely for another complete period before a second explosion takes place.

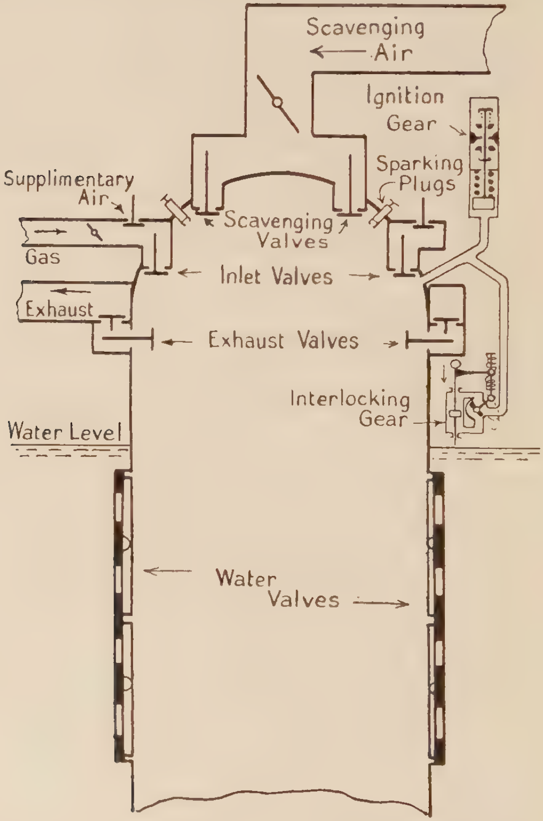


FIG. 13.—Cylinder Head of Humphrey Gas Pump.

The means adopted to carry this out are quite original and worth studying in some detail. The cylinder head is surrounded by three belts of automatic mushroom valves. At the top are placed eight air scavenging valves, which open inwards and are normally held closed by light springs. Next come eight gas (and air) inlet valves, which are also held closed by light springs. The supplementary air valves do not open direct into the combustion space, but are placed in the gas duct leading to the inlet valves. Below these are the exhaust valves, sixteen in number, which are automatically held *open* by springs. They each have a non-return valve behind them which prevents any reverse in the exhaust pipe. Immersed below the normal level of the water are a large number of water flap valves which open inwards and are closed again by springs when the pressure is the same on both their sides. The scavenging air and the gas inlets are both fitted with butterfly throttle valves for hand control.

In order to carry out the cycle of events with these valves, an interlocking device is necessary, which prevents the various valves from opening automatically when not required. This device has to operate over every two strokes and it is worked by the gear seen below on the right-hand side. This gear consists of a small cylinder with a double-acting piston inside. The piston rod of this piston connects to the interlocking device so that when the piston is down one set of valves is locked, and when it is up the others are held closed. The piston is moved by the water itself as it reaches the top of each return stroke. The water can then flow into the pipe between the inlet and the exhaust valve and is passed to the top or to the bottom of the operating cylinder, according to the position of the small oscillating valve at the cylinder side. This oscillating valve is turned over by the movement of the little piston, a lever from it being attached, as shown, to the top and the bottom of two springs. The other ends of these springs are attached to the piston rod, but they are so weak that they do not pull the oscillating valve over until the water pressure is off. The gear is then set for another stroke as soon as the water returns to the level of the intake pipe.

Ignition is by a ring of sparking plugs which are also actuated by the water at the top of each return stroke. The gear can be seen above on the right-hand side. A small piston is pressed up against a strong spring, which returns it as soon as the water pressure is off. The piston rod carries two conical discs which are connected to it with light springs. The upper disc rests on a seating when the piston is at the bottom of its stroke. As the piston rises the light spring keeps this disc on its seat until the lower disc also beds on to a seating prepared for it underneath. The ignition coils are electrically connected up in such a way that a current flows only when both discs are in contact with their seatings together. The gear functions every other stroke, but the mixture in the combustion space is too weak to explode at the end of the exhaust stroke.

Cycle of Operations of the Humphrey Gas Pump.—The four strokes of this Humphrey gas pump are shown in Fig. 14. It is convenient, in this instance, to commence with the firing stroke, Fig. 14 (*a*). The water has swung back up the cylinder from the preceding cycle and compressed the gaseous mixture in the combustion space. The interlocking gear is operating on the gas-inlet valves, whilst the air scavenge is held closed by the compression as well as by its spring, and the exhaust valves are closed by the pressure from the water. Explosion takes place at constant volume and the gases subsequently expand to slightly below atmospheric pressure. In this, the cycle differs from all previous ones already described, in which expansions stop short at some pressure in excess of that due to the atmosphere. The nearest approach to it is the Atkinson cycle of Fig. 9, which has an extended expansion, but did not, in the actual engines, reach down to the atmospheric pressure line. The scavenging air valves open as the pressure falls below that of the atmosphere. The exhaust valves have opened as soon as the water leaves them, but the non-return valves behind prevent any exhaust products from returning into the cylinder. The water valves also open towards the end of this stroke and water pours in which supplements the original water level for the next stroke.

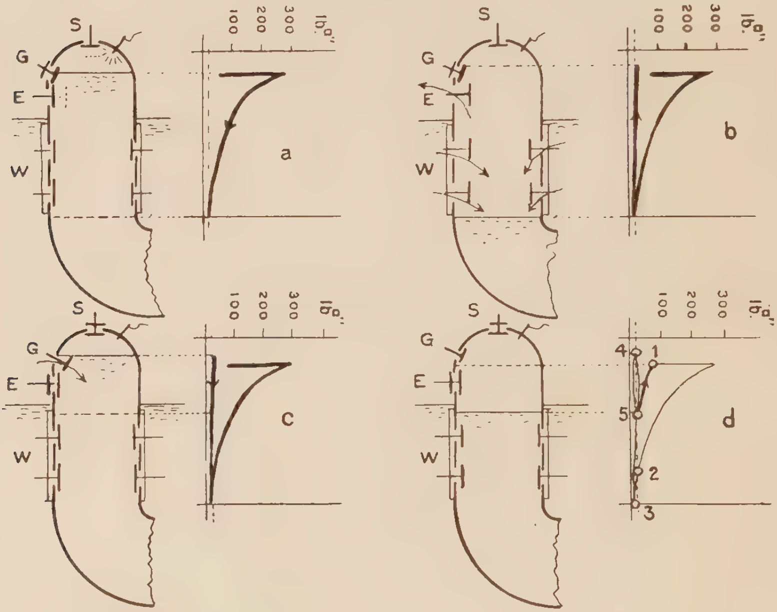


FIG. 14.—Cycle of Operations of Humphrey Gas Pump.

(*a*) Ignition and expansion.
(*b*) Exhaust.
(*c*) Suction.
(*d*) Compression.

The swinging column of water is gradually brought to rest under the influence of these different factors and the water starts rising in the chamber. This produces the exhaust stroke, Fig. 14 (*b*), during which period the gas-inlet valves are still locked in a closed position, and the air-scavenge valves are held closed by themselves. As the water passes the exhaust valves, it closes them automatically and immediately afterwards changes over the interlocking gear, which now operates on the exhaust valves and on the air scavenge, so that neither of these sets of valves can alter again during the next two strokes. The gas-inlet valves, on the other hand, have been released from their lock and are free to function on the suction stroke which follows. The swinging column of water is brought to rest against the cushioning effect of the trapped air, but as more water has been added, the level is slightly higher than on the previous return stroke.

The suction stroke follows (Fig. 14, *c*), drawing in gas and supplementary air through the gas-inlet valves. As there is no impulse, this stroke is much shorter than the previous outstroke.

The last stroke (Fig. 14, *d*) compresses the gaseous mixture as shown, all the valves being closed and the exhaust valves being still locked. A change over of the interlocking gear takes place at the end of this stroke, and the cycle repeats itself.

The operation of the pump is necessarily slow, the large pumps at Chingford making about 9 complete cycles per minute, and this necessitates a relatively large cylinder for the power developed; but the cycle on which the pump works can be shown to be more efficient in theory than the Otto four-stroke cycle, in which exhaust takes place at constant volume, instead of at constant pressure.

The following schedule may be of assistance in understanding the operation of the various valves. The numbers correspond to those on the indicator card in Fig. 14 (*d*).

L=locked. O=open. C=closed.

	Exhaust.	Gas.	Scavenge.
1 to 2 Ignition and expansion . . .	C	L	C
2 to 3 Scavenge	O	L	O
3 to 4 Exhaust	O	L	C
Interlocking gear changes over.			
4 to 5 Suction	L	O	L
5 to 1 Compression	L	C	L
Interlocking gear changes over.			

CHAPTER III

OIL ENGINES AND THEIR CYCLES OF OPERATION

General.—The internal-combustion engine which works with oil as a fuel is nearly as old as its prototype the gas engine. Most of the early inventors, such as Lenoir or Otto, worded their patent specifications in such a way as to show that they contemplated using oil vapour as well as gas for their combustible mixture, and it is difficult, at this date, to say when oil vapour was first successfully tried. The Brayton petroleum engine, which was developed in America in 1873, is generally considered to be the pioneer (see p. 16), but this engine was first designed to work on gas. The credit of producing the first reliable engine to work with crude oil on the Otto cycle is usually attributed to the brothers W. D. and S. Priestman (p. 38), who patented their well-known engine in 1886. Just as the steam engine influenced the design of early gas engines, so did the gas engine influence the design of these early forms of oil engines. The Priestman engine was to all intents and purposes a gas engine which worked on a mixture of oil vapour and air, and the vaporisation took place outside the working cylinder.

In 1890, H. Akroyd-Stuart and C. R. Binney took out a joint patent (No. 7146 of 1890), which, for the first time, introduced the idea of vaporising the oil in a chamber in contact with the working cylinder, and their ideas were embodied in the Hornsby-Akroyd oil engine which also worked on the Otto cycle. It was not until R. Diesel, about the year 1894, suggested the idea of compressing air alone in the working cylinder and injecting the fuel at the end of compression, that a radical departure was made from the Otto cycle, which enabled higher thermal efficiencies to be obtained than is possible in a gas engine.

In the gas engine a combustible mixture is always compressed, and, as rapid compression of any gas raises its temperature, a pressure could be reached which would cause spontaneous ignition. This definitely limits the amount of compression, or compression ratio as it is called, which is possible in such engines.

The same limitations, of course, are present in engines using oil vapour, such as petrol engines, which compress an inflammable mixture. Both theory and practice show that, for nearly all types of internal-combustion engine, the higher the ratio of compression the higher the thermal efficiency will be, so that when Diesel introduced the idea of compressing a non-inflammable gas such as air and injecting the fuel afterwards, he opened out great possibilities for the oil engine which gave it a distinctive character of its own. It is rather curious that no serious attempt has been made in recent times to apply this principle to the gas engine. Experimental research would be necessary to find out whether such a thing were practicable, but there appears to be, on paper, no inherent difficulty to the compression and injection of a pure gas into a gas-engine cylinder at the end of its compression stroke. The method would naturally lend itself best to a gas with a high calorific value, such as town gas, because the gas compressor would be so much the smaller, but it might even be worth considering for producer gas.

In this connection, it is interesting to note that Mr. Dugald Clerk, as he then was, described[1] a two-stroke engine which introduced compressed gas in the compression space near the end of the compression stroke. Two such engines were made as far back as 1888, but the type was not proceeded with because at that period greater economy could be obtained from the constant-volume explosion engine.

Classification.—The multiplicity of types which the modern oil engine has assumed, and the number of variables involved, have made it almost impossible to find a method of classification which would be generally recognised. The various types of fuel used, the various methods of forming the working substance, the different compressions and maximum pressures reached, the different thermodynamic cycles on which the various engines work, the different names used in industry, and the very different mechanical constructions which have been evolved, all offer different methods for classification. But as these factors are, to a large extent, interdependent, no one method would really identify the engine type.

The following is the complete report of the Committee of the Institution of Mechanical Engineers which considered the matter of oil-engine nomenclature under the chairmanship of Capt. H. Riall Sankey.[2] It is reproduced here by the kind permission of their Council.

> 1. The Committee has given consideration to various suggestions and schemes of nomenclature emanating from the Chairman, individual members, and from its sub-committee; to correspondence and pamphlets addressed to it by Mr. Herbert Akroyd-Stuart; to dossiers submitted by the Admiralty and the Department of Scientific and Industrial Research; to representations from the British Engineering Standards Association; and to replies to inquiries, made on its behalf by Mr. Pendred, as to the existence or otherwise of approved nomenclature of oil engines by engineering institutions in France and in the United States of America.
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> 2. The Committee is of the opinion that the modern oil engine,

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[1] In the discussion on "The Diesel Oil Engine," by R. Diesel, Proc. Inst. Mech. Engineers, 1912, Parts I. and II., p. 232 *et seq.* Indicator cards from these engines, p. 234 (*ibid.*), show that such an engine worked on what is now known as the dual-combustion cycle.

[2] Proc. Inst. Mech. Eng., Nov. 1922, p. 1114.

as distinct from the automobile type of liquid-fuel engine, has been evolved mainly from the principles enunciated by Mr. Herbert Akroyd-Stuart in his Patent No. 7146 of 8th May, 1890, and, as embodied in the engine constructed at Bletchley (Buckinghamshire) and tested by Professor William Robinson, a member of the Committee, in February, 1891. Professor Robinson referred to this engine in detail in a paper, "The Uses of Petroleum in Prime Movers," read on 29th April, 1891, and subsequently published in the *Journal* of the Society of Arts. This "Akroyd" engine is the prototype of those in which the liquid fuel is introduced into a compressed or partially compressed charge of air, and which do not need an extraneous source of ignition. In this respect the "Akroyd" engine anticipated the engine subsequently evolved in Augsburg, Germany, by the Maschinenfabrik Augsburg Nürnberg from the original proposals of Rudolph Diesel in 1893 and described by H. Ade. Clark in his paper "The Diesel Engine" read before the Institution at the meeting in Leeds in July, 1903. It may thus be said that the pioneer work of Herbert Akroyd-Stuart has been adversely cloaked by nomenclature.

3. In the development of modern oil engines, departures have been made both from the "Akroyd" and the "Diesel" original principles, and to designate these by inventors' names would be misleading and depreciative of the work of those to whose efforts the recent successes achieved have been due. The various types of engines merge so intricately that arbitrary grouping for the purposes of nomenclature would be unsatisfactory. For commercial exploitation, trade names cannot be dispensed with, and, after all, these are usually sufficiently distinctive for general use.

4. The Committee considers that the term "Semi-Diesel" as applied to oil engines is subversive of the fact that this type of oil engine is a British production and an evolution from the "Akroyd" engine.

5. After careful consideration, the Committee has been forced to the conclusion that an extended classification would be of little value, and that it is improbable that any system of nomenclature could be devised to cover all divergent points of detail and to meet practical requirements.

Such a report is not the less valuable because it happens to be negative. It shows that the developments of the oil engine at the present time are so numerous and diverse that it is not yet possible to record that any specific type is standardised or incapable of improvement. A time, no doubt, will come, as it did in the case of the steam engine, when this will be possible, but meanwhile the study of the application of oil fuel to internal-combustion engineering remains in a very interesting and important position.

All that can properly be done at present is to arrange the various developments of oil engines into a series of groups.

The Grouping of Different Types of Oil Engines.—In the *first group* may be considered all those oil engines which function like a gas engine; that is to say, engines which employ some form of vaporiser to produce a mixture of oil vapour and air, and which draw such a mixture into the working cylinder on the suction stroke. If the engine is working on the two-stroke cycle this mixture would be introduced towards the end of the working stroke. In both cases an essential feature of this group lies in the fact that the working substance is compressed in the cylinder and is liable to pre-ignition if this compression is carried too far. Also some form of ignition device is necessary.

Experience has taught that the time element plays an important part in the design of engines of this kind. The result is that high-speed oil engines, such as those used for mechanical transport or aviation, although similar in principle to the first group, really form a branch of study by themselves. This difference is accentuated by the fact that the light oils, such as petrol or alcohol, which are used in such engines, volatilise so readily at normal temperatures and pressures that no vaporiser with its supplementary heating is required, and an apparatus known as a carburettor is used instead. Both these branches of the first group work on the constant-volume cycle.

In the *second group* oil is injected near the end of the compression stroke, and the danger of pre-ignition is avoided unless the fuel valve leaks. The vaporiser is modified to form a " hot-bulb " on the cylinder itself, and the oil is usually broken up as it enters the compression space by specially designed nozzles. When starting, this bulb is heated on the outside with a hand-lamp, but after the engine has warmed up, it is found that no extraneous ignition device is required. The compression in these engines may be higher than in the first group, but in most cases they work on the constant-volume cycle. On account of the fact that only air is compressed, engines in this group are frequently referred to as " Semi-Diesels," but the best generic name is probably *Hot-bulb oil engines*.

Engines in this group may be distinguished from the Diesel engine proper by the type of indicator card which is obtained. The Semi-Diesel works on the constant-volume cycle, whilst the Diesel has a distinctive card of its own which is characterised, amongst other things, by a flat top, denoting constant-pressure combustion.

The *third group* consists of the most recent development in oil engines and has not yet received a distinctive name. They differ from the second group in that no hot-bulb is used, but are similar in that air only is compressed. The essential feature of this group is a mechanically operated oil sprayer, which is powerful enough to atomise the oil into such a fine spray that it intermingles with the air in the compression space sufficiently to ignite without the help of a vaporiser. The compression of the air is carried just far enough to ensure ignition without any other ignition device, and burning takes place mostly at constant volume, though to a small degree at constant pressure as well.

Various names which have been applied to engines of this group are: *cold-starting* engines, from the fact that they do not require a hand-lamp to start them, as in the case of hot-bulb engines; *solid-injection*, *airless-injection*, or *mechanical-injection* oil engines, to distinguish them from the better known Diesel oil engine of the fourth group, which atomises its oil by means of an air blast; and *sprayer* oil engines, from the fact that the oil sprayer is their most essential and distinctive feature, in that it fulfils more functions than in any other group.

The *fourth group* includes the distinctive engine with which the name of Diesel is associated. In this engine, air is compressed up to the maximum pressure of the cycle and the oil fuel is injected by means of an air blast and burns at approximately constant pressure. No ignition device is necessary, the heat of compression in the air being ample for the purpose. From the fact that the time of combustion is slow compared to the combustion in the constant-volume cycle, these engines are occasionally called slow-combustion oil engines. An air compressor is a necessary adjunct to this type of engine and the mechanical efficiency must therefore be lower than would otherwise be the case; but for a long time these engines had the highest thermal efficiency of any type of heat engine, which more than counter-balanced the power required for air compression.

The engines in all these groups can be made to operate on either the four-stroke or the two-stroke cycle, and there is, at the present day, a wide diversity in their mechanical construction.

The Priestman Oil Engine.—This engine may be considered to be the first successful oil engine which was designed expressly to run on heavy oils. It was patented by W. D. and S. Priestman in 1886 (No. 1394), and commercial engines of the type were in use in 1888.

A diagram of this engine is shown in Fig. 15, but, as it worked on the four-stroke Otto cycle, it is not necessary to detail out the separate strokes. The cylinder was arranged horizontally with a trunk piston connected to a crankshaft and a heavy flywheel. The inner liner was amply water-jacketed with the inlet and exhaust valves placed at one end. The inlet valve was automatic in its action; that is to say, it was held on its seating by a spring and opened only during the suction stroke of the cycle. The exhaust valve was beneath the inlet valve and was opened by an eccentric, which was driven by a lay shaft running at half the speed of the engine. This valve was timed to open only during the exhaust stroke of the cycle.

The sparking plug was placed at the side of the compression space, as shown, and was worked by means of a battery and an induction coil. The circuit was completed at the proper moment by a contact finger, which was carried on the eccentric rod. An air pump was mounted alongside the working cylinder and driven at half the speed of the main piston by the same eccentric which was working the exhaust valve and timing the spark. This pump supplied air at about 8 lb. per sq. in. above atmosphere to a supply tank, which contained the fuel oil. There was a relief valve on this tank (not shown) which kept the air pressure constant inside. A small hand-pump was also mounted on the fuel tank to pump up the pressure when starting.

A vaporiser was placed underneath the main cylinder in the engine bedplate and the exhaust gases were arranged to flow round it after leaving the engine. A lamp was connected up to the fuel tank, and used to warm up the vaporiser when starting up. The nozzle, which can be seen in the detail, had a central passage for the oil and a surrounding space for air. Both the oil and the air came from the fuel tank, but the air was diverted round to blow into the oil as it issued from the nozzle. The result was to get a fine spray on much the same principle as the scent sprays which are used by hairdressers. Extra air could be drawn in as shown without passing through the nozzle. The supply of this air was controlled by a butterfly-wing valve. An explosive mixture of oil vapour and air was formed in the vaporiser and drawn into the main cylinder through the automatic inlet valve on the suction stroke. The mixture

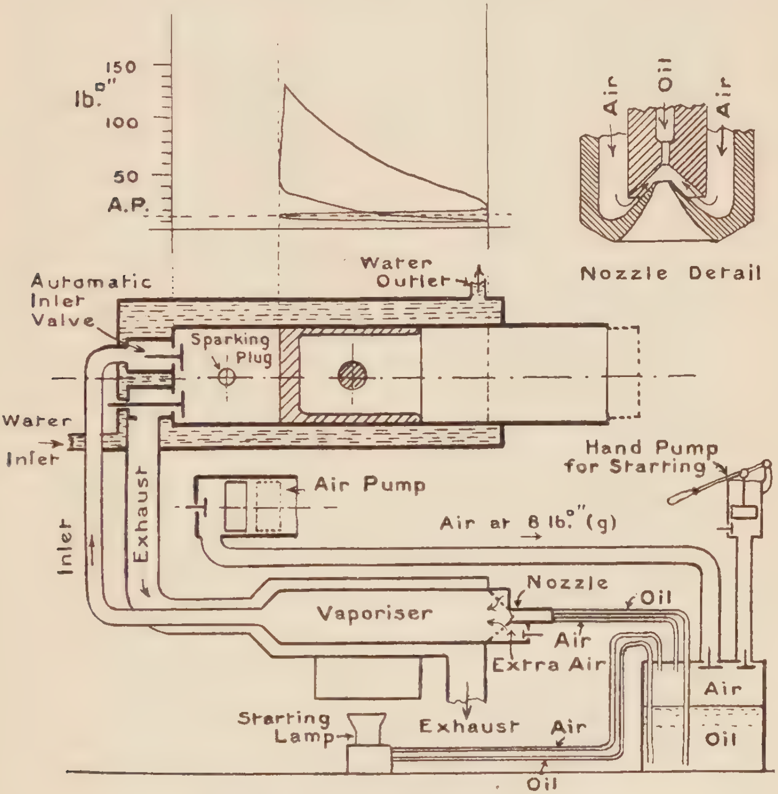

FIG. 15.—Priestman Oil Engine.

was then compressed, fired, expanded, and exhausted as in a gas engine.

A 5 H.P. Priestman oil engine was tested by Prof. W. C. Unwin at the Royal Agricultural Society's Show at Plymouth, in 1889, of which full details are given in the Proc. Inst. Civil Eng., vol. cix. p. 1 (1892). The compression ratio was 3 to 1, and the brake thermal efficiency was 15·5 per cent. when using "Daylight" oil with a lower calorific value of 19,700 B.Th.U. per lb. and a specific gravity at 60° F. of 0·79. The consumption was therefore 0·84 lb. of oil per B.H.P. per hour. The ratio of air to fuel as measured by means of an anemometer was 33·4 to 1. This was exclusive of the air from the fuel tank which was used to form the spray, and which Prof. Unwin estimated amounted to less than one-thirtieth of the amount measured.

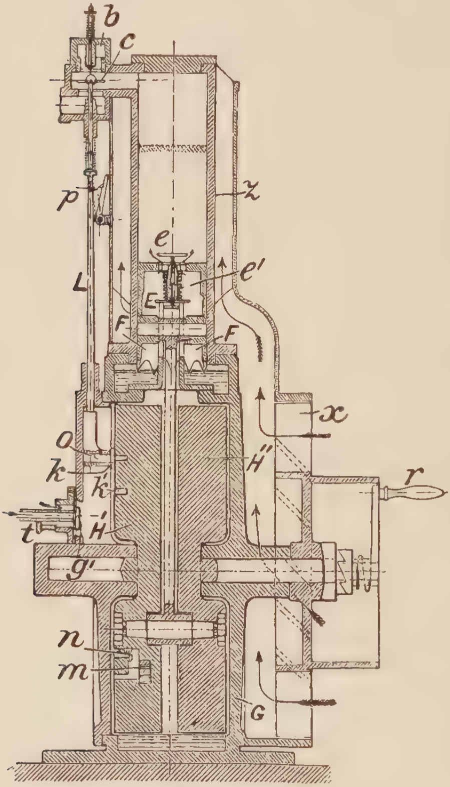

Fig. 16.—Daimler Engine of 1885.

The Daimler Engine of 1885.—It is not generally realised that, about the same time that the brothers Priestman were developing their oil engine, an Austrian, G. Daimler, evolved the germ of the petrol engine. His English patent specification is so interesting that permission has been obtained to reproduce it here in its abridged form with its accompanying diagram (Fig. 16).

4315. Daimler, G., April 7, 1885.

Gas Engines.—A complete cycle is performed in two revolutions. The flywheel H′, H″ carrying the crank-pin is enclosed in a chamber G open to the lower end of the cylinder. As the piston approaches the bottom of its first down-stroke after an ignition, the exhaust valve *c* is opened and allows some of the products to escape; at the same time the projection F compresses the spring *e*′ and leaves the valve *e* in the piston E free to open and admit air or combustible mixture which on the previous stroke had been

> compressed in the chamber G into the cylinder; on the up-stroke more products are expelled from the cylinder. During the next down-stroke the main charge, consisting of a mixture of combustible gas or vapour and air, is drawn in through the valve *b*; when the bottom is nearly reached the valve *e* is again freed by the projection F and a further charge admitted from the chamber G; on the up-stroke compression takes place and then ignition by a hot tube, a premature ignition being avoided by weakening the charge near the hot tube by closing the gas valve towards the end of the piston's second down-stroke by a tappet on the rod L. The exhaust valve *c* is actuated by a cam *k*, *k'* on the flywheel through the pin *o* and rod L. When the speed is excessive a piece *m* pivoted at one end to the wheel tends to fly outwards, and actuates through the other end a cam switch *n* which guides the block on the pin *o* a second time round into the concentric groove *k'* of the cam and keeps the exhaust valve closed during another revolution. The cycle may be varied by dispensing with either the first or second charge admitted through the piston by keeping the inlet valve *g'* open at suitable parts of the stroke by a tappet on the end of the valve rod L. The engine is started by keeping open the exhaust valve *c* by the prop *p*, and the inlet valve *g'* by the screw *t*, and rotating the crankshaft by the handle *r*. The cylinder is cooled by air forced through the jacket *z* by the fan *x*.[1]

It will be noted that Daimler foresaw the possibility of a mixture of oil vapour and air, although his engine was primarily intended for gas. Some form of carburettor would be required to obtain such a mixture, but it is interesting to note how many points in the design of modern petrol engines are foreshadowed in this specification.

The Hornsby-Akroyd Oil Engine of 1890.—It has already been noted that this engine represents the first successful attempt to evolve a heavy-oil engine proper as opposed to the work of previous designers who attempted to adapt the principles of the gas engine to an engine using vaporised oils. It may be considered as the true forerunner of most modern oil engines, in that it was the first design to inject oil direct into a chamber which formed part of the compression space.

A diagram (in plan) of this engine can be seen in Fig. 17. The original engines worked on the four-stroke Otto cycle, though they were subsequently designed for the two-stroke cycle as well. The cylinder had a flat end and it was nearly filled by the piston at the end of the inward stroke, but a separate chamber, known as the vaporiser, was in direct communication with the cylinder by means of a "bottle neck." A second outlet from the cylinder proper led to a chest which contained an exhaust valve and an inlet valve for air, both of which were of the mushroom type and were actuated by rocker levers, which passed under the engine and engaged with cams placed on a half-motion lay shaft. The cylinder, exhaust valve, and part of the vaporiser were well water-jacketed, but the further end of the vaporiser was left bare. When the engine was being started, a powerful blow-lamp heated up this free end, and a casing-cover placed round it helped to retain this heat. At the proper time, crude oil was injected through a nozzle into this vaporiser as shown. The oil was pumped in by a small plunger pump which was fitted with a bye-pass, operated by a governor to control the quantity delivered according to the speed of the engine.

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[1] From Abridgements of Specifications (1896), class 7. Air and gas engines. Period, A.D. 1884–1888. By kind permission of the Controller of H.M. Stationery Office.

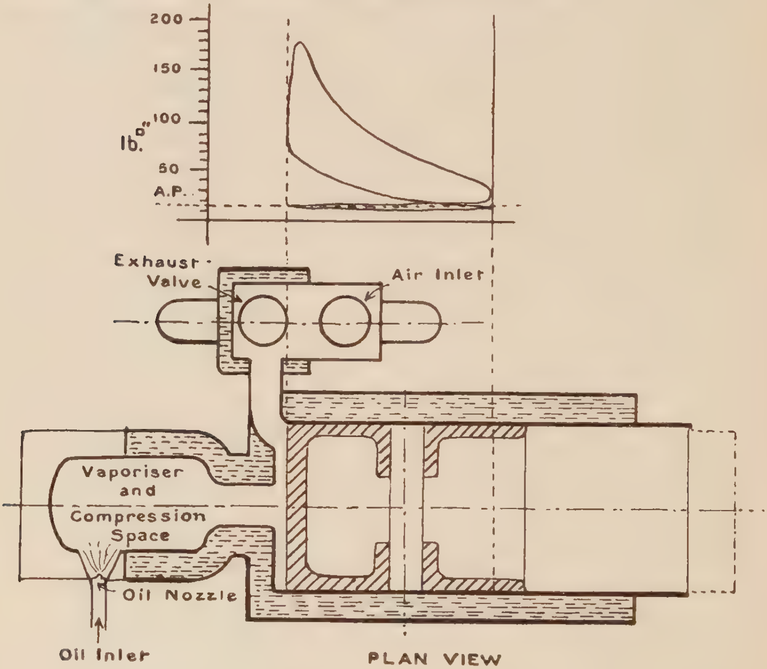

FIG. 17.—Hornsby-Akroyd Oil Engine of 1890.

When it was running, the engine functioned in the following manner. The oil fuel was injected at, or near, the commencement of the suction stroke, and volatilised by the hot walls. The vaporiser at this period of the cycle was filled with products of combustion from the last working stroke, and there was not sufficient fresh air present to form an explosive mixture. Immediately after the nozzle had functioned, the air inlet valve was opened and fresh air was drawn into the cylinder proper for the remainder of this outward stroke. The next return stroke compressed this air and drove it through the " bottle neck " at turbulent velocity to intermingle with the oil vapour in the vaporiser. The volume of the vaporiser and the ratio of compression were so proportioned that the mixture became explosive at the end of the compression stroke and was fired by the heat of the walls. Combustion took place at constant volume, but it was necessary to alter the compression ratio by means of plates in the big end of the connecting rod, or sometimes by fitting another vaporiser, in order to get the best results with different grades of oil. The working and exhaust strokes were similar to those already described for the Otto cycle, but it should be noted that the exhaust gases were expelled through the same port and valve chest as was used for the fresh air inlet and that consequently this part of the engine was subject to considerable temperature fluctuation.

This engine was a great success, and many of them are still running. Sir Dugald Clerk gives details[1] of the first trials which were made by Professor Capper on an 8 B.H.P. engine at the Royal Agricultural show in Cambridge. The cylinder diameter was 10 in. and the stroke 15 in. The speed was 240 r.p.m., and the mechanical efficiency 83 per cent. The oil consumption was just under 1·0 lb. per B.H.P. per hour. The oil used was Russoline, with a lower calorific value of 18,550 B.Th.U. per lb. and a specific gravity of 0·82. The mean effective pressure was 29 lb. per sq. in. and the maximum explosion pressure about 127 lb. per sq. in. abs. The ratio of compression was 2·875. This result was considerably improved upon in course of time. A test in 1908 by Professor Robinson, for instance, on a 32 B.H.P. engine using Russoline oil, showed a consumption of 0·613 lb. of oil per B.H.P. per hour, and a brake thermal efficiency of 22·6 per cent. It was the first engine to be made in large sizes, as much as 200 I.H.P. being developed in a single horizontal cylinder.

The Diesel Oil Engine.—The next step in the evolution of the oil engine led to a type which proved to be, until quite recently, the most efficient way of converting heat into work. This engine was the outcome of original thought and experiment by a German, Dr. Rudolf Diesel. Instead of trying to follow and improve on the work of his predecessors, Diesel considered the various cycles which are possible for use in heat engines, and eventually evolved a new type of cycle which is known by his name. His theoretical considerations are set out in a brochure[2] which was translated into English in 1893 under the title of "The Rational Heat Motor." His originality of thought may be gauged from the fact that he first proposed to design an oil engine to work on the Carnot cycle, which, in theory, is the most efficient of all heat-engine cycles, and only abandoned it when the design showed that the engine would prove to be too heavy to be practical. The first experimental engine actually constructed was made in Augsburg in the year 1893, and here again the courage of this inventor is apparent, as it was originally designed to run on such a coarse fuel as coal dust, and a thermal efficiency of 70 per cent. was expected. The principle was sound, but

[1] Dugald Clerk, "The Gas and Oil Engine," 1902 ed. p. 425.

[2] "Theorie und Konstruction eines rationellen Wärmemotors zum Ersatz der Dampfmaschinen und der heute bekannten Verbrennungsmotoren," von R. Diesel, Berlin, 1893. There is a good summary of the contents of this pamphlet in the *Zeitschrift des Vereines deutscher Ingenieure*, vol. 37, 1893, p. 291, and copies of the English translation may be seen in the libraries of the Institutions of Civil and of Mechanical Engineers, London.

difficulties of pressures and temperatures arose, and crude oil was substituted. After four years, a reliable machine was produced,[1] and the engine made headway as a commercial proposition. Mechanical troubles still arose, but the progress and development of the Diesel engine for the next fourteen years is very clearly set out in a paper read by the inventor before the Institution of Mechanical Engineers.[2] The engine took its place as the most important type of oil engine, particularly for larger sizes than had hitherto seemed possible, and it was, and is, manufactured by numerous firms in Great Britain, on the Continent, and in America. A society, the Diesel Engine Users' Association, was formed for those who were concerned with the running of Diesel-engine plants.

The essential difference between the Diesel oil engine and its forerunners lies in the fact that air only is compressed to a pressure which ensures its reaching a sufficiently high temperature to ignite any fuel which can be introduced at the end of the compression stroke so as to form a combustible mixture; no other ignition device is required. Diesel used an air-blast to introduce his fuel near the end of the compression stroke. It soon became apparent that this method, besides reaching a higher thermal efficiency than had ever been previously attained in any form of heat engine, enabled all kinds of coarse fuel, such as crude oils, to be used without previous vaporisation.

The Diesel principle can be applied to engines working on the four-stroke or on the two-stroke cycle, and cylinders may be placed vertically or horizontally. The method can also be used with the opposed-piston type similar to the Oechelhauser gas engine, or with double-acting cylinders like a steam engine. The essential parts of the simplest form of Diesel engine can be seen in the diagram of Fig. 18.

This engine works on the four-stroke cycle and the cylinder is placed vertically. A trunk piston is moved up and down in a cast-iron cylinder liner which is surrounded by an ample water-jacket. The cylinder head is separate, and has to be very carefully constructed to withstand not only the maximum pressures inside the cylinder but also the very considerable temperature stresses which may be reached with this type of cycle. It is usually made of a special cast-iron mixture, though cast steel is occasionally used. It contains the air-inlet valve, the fuel-injection valve and the exhaust valve (or valves) as shown, as well as a smaller air-starting valve which is placed in the opposite plane, in front of the fuel valve.

The air-inlet and exhaust valves are opened by rocker levers from a half-motion cam shaft, and are closed by strong springs. The needle of the fuel-injection valve is also operated by a lever and cam, and it usually has a constant period of opening. Control is obtained by varying the quantity of fuel supplied by the fuel pump per stroke. This arrangement may be seen on the left-hand side.

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[1] For the first publication of Diesel's results, see "Diesel's rationeller Wärmemotor," von R. Diesel, *Zeitschrift des Vereines deutscher Ingenieure*, vol. 41, 1897, pp. 785 and 845.

[2] "The Diesel Oil Engine," by R. Diesel, Proc. Inst. Mech. Eng., 1912, Parts I. and II., p. 179.

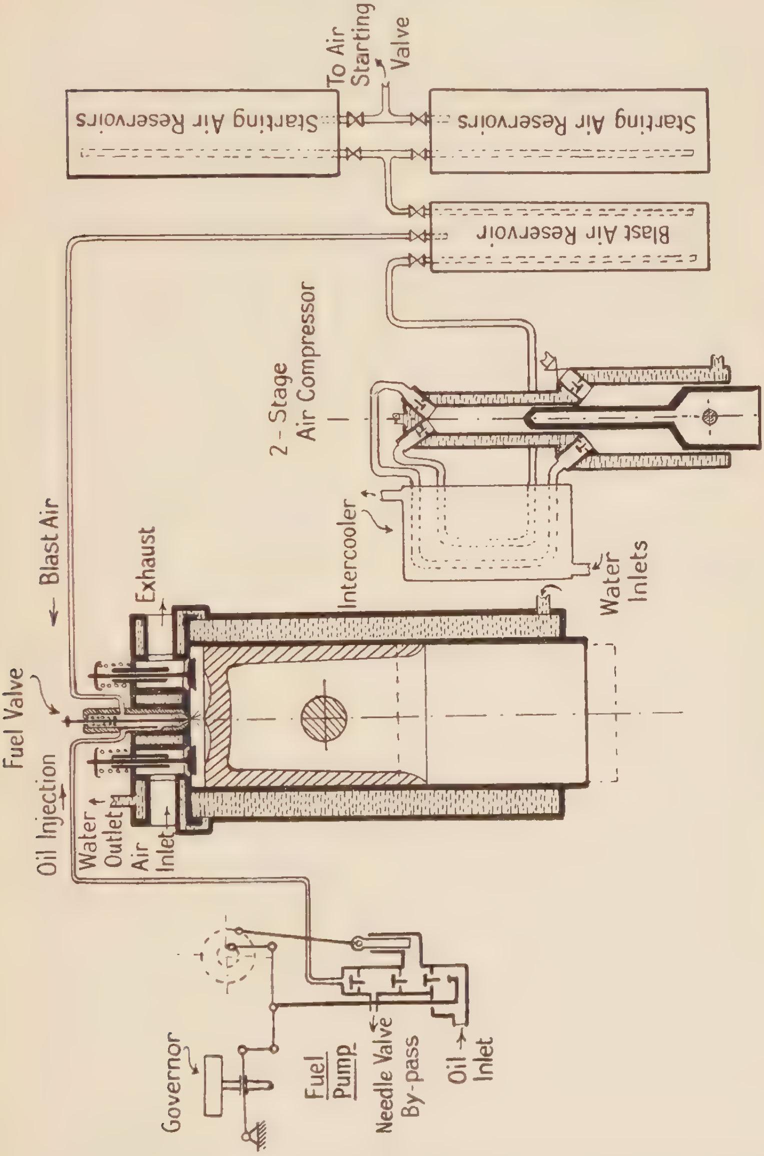

FIG. 18.—Essential Parts of Diesel's Four-Stroke Cycle Oil Engine.

A small steel plunger is worked either by an eccentric or from the end of the half-motion shaft and pumps a definite quantity of oil per stroke.

The suction valve of this pump is held open by means of a hooked stop which is also driven by the half-motion shaft. If the speed of the engine becomes excessive, the governor, as shown, raises the pivot of the lever which actuates this stop, the suction valve is held open for a longer period than it would have normally and no oil is delivered to the fuel valve until the suction valve returns to its seat.

Two delivery valves are usual, automatic in action and arranged as shown in series. This enables the attendant at any time to ascertain whether the pump is working satisfactorily by opening a small needle bye-pass valve and noting the spray of oil which is ejected. The capacity of these pumps is usually between four and five times the normal fuel requirements, and they require very careful workmanship as they have to deliver at a pressure in the fuel valve (say 1000 lb. per sq. in.) which is in excess of that of the blast air. The oil is usually introduced into the nozzle just above a series of baffle plates, which help to break it up into very finely divided particles when the air blast is on. This air is delivered into the fuel valve at about 800 lb. per sq. in. and requires an air-compressor which may be two- or three-stage, and which is driven either from the end of the main crankshaft or by rocking levers attached to some point of the reciprocating parts. The diagram of a common form of two-stage air-compressor is shown on the right. The first stage is below, air being drawn into the water-cooled cylinder through the valve shown on the right and compressed to about 120 lb. per sq. in. It is then passed through a water-cooled coil, shown diagrammatically in dotted lines, to the second stage, where it is raised to between 900 and 1000 lb. per sq. in.

From the compressor the air is again passed through coils in either the same or in a separate cooler and stored in a strongly made air bottle marked blast-air reservoir. When the engine is running, this air bottle is in direct connection with the fuel valve and the pressure can be regulated, if necessary, by the valve shown on the suction of the first stage of the compressor. A number of pressure-gauges (not shown) are mounted in convenient positions for the guidance of the attendant. Advantage is taken of the presence of an air compressor to collect and store compressed air at about 700 lb. per sq. in. for starting purposes. Two starting-air reservoirs are shown but only one is required at a time.

It will be noted, from the above brief description, that the Diesel engine is a more complicated unit, even in its simplest form, than the other types of oil engine, and although the thermal efficiency is very high, the mechanical efficiency is in the neighbourhood of 75 per cent., chiefly because of the power required to drive the air-compressor.

The Diesel engine is made with trunk pistons and also with crossheads similar to high-speed vertical steam engines. Larger powers are obtained by multiplying cylinders side by side. Engines of 2500 B.H.P. are now common for marine work.[1]

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[1] " Clyde Marine Oil Engines," by A. L. Mellanby, Proc. Inst. Mech. Eng., 1923, Vol. II., p. 695.

The fuel consumption may be as low as 0·38 lb. of oil per B.H.P. per hour, with oil of a lower calorific value of 18,000 B.Th.U. per lb., and the corresponding absolute thermal efficiency 37·2 per cent.

The Cycle of Operations of the Four-stroke Diesel Engine.—In Fig. 19 are shown the four strokes of a Diesel engine, together with the two of the air-compressor. These latter are repeated during the cycle of the main engine. In the top left-hand diagram, the suction stroke is shown, and the approximate time during which the air-inlet valve remains open may be seen from the length of the arc marked 1 and 2. On the first return stroke, seen in the top right-hand diagram, air only is compressed to between 470 and

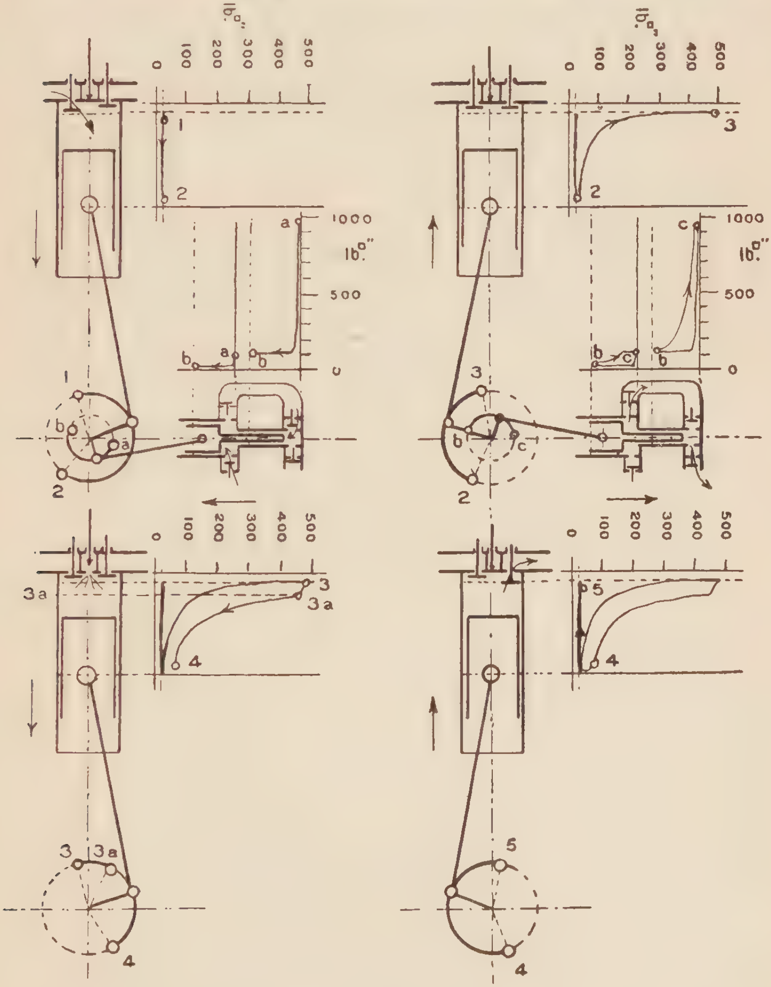

FIG. 19.—Cycle of Operations of a Four-Stroke Cycle Diesel Oil Engine.

500 lb. per sq. in. The compression ratio of the Diesel engine is about 14:1, and compression takes place over the crank-pin arc marked 2 to 3. The combustion and expansion stroke is shown in the bottom left-hand corner. Fuel is injected over the period marked 3 to 3*a*, and burns approximately at constant pressure. The fuel valve closes at 3*a* and expansion takes place throughout the remainder of this stroke. At the point 4, the exhaust valve opens and exhaust takes place down to atmosphere at approximately constant volume, as shown in the remaining diagram. The products of combustion are then expelled by the upward movement of the piston. There is usually a small amount of overlap between the opening of the air-inlet valves and the closing of the exhaust valve so as to clear the compression space as much as possible of exhaust gas.

The corresponding diagrams of pressure changes in a two-stage air-compressor are also shown. For clarity they are drawn at right angles to the main axis, though in practice the two cylinder axes are usually parallel. The two pistons are interlocked, suction takes place from *a* to *b* and compression from *b* to *c*. In the first stage, the suction is from atmospheric air and the delivery to the second stage, through an intercooler, may be about 120 lb. per sq. in. (abs.). In the second stage, suction starts at about that pressure and delivery to the air bottles is at about 950 lb. per sq. in.

The relative size of these two diagrams is rather deceptive, because they had to be plotted on a stroke basis, whereas they should really be studied on a volume basis. Actually the two cylinders are proportioned to do approximately equal amounts of work.

Two-stroke Diesel Oil Engines.—The two-stroke Diesel engine requires, in addition to the fuel-oil pump, air-compressor, coolers, and air reservoirs shown in Fig. 18, some form of scavenging pump. It is practically never made with crank-case compression scavenge such as will be described later for hot-bulb oil engines (Fig. 22, p. 52).

There are, however, four distinct ways in which the scavenge of the cylinder is effected. These are shown diagrammatically in Fig. 20. In each case a single-acting scavenging pump is indicated, but in practice, particularly when two or more cylinders are fitted, these pumps are made double-acting, though one such pump often serves four cylinders.

In all four cases, exhaust valves are absent and the products of combustion leave the cylinder through ports which are uncovered by the piston near the end of the stroke. The first method is known as valve scavenge. Two or four scavenging-air inlet valves are placed in the cylinder head and mechanically operated by rocking levers to admit fresh air from the scavenging pump at about 5 lb. per sq. in. above atmosphere. Compression starts at this pressure when the piston overruns the exhaust ports on its return stroke.

A second method, known as port scavenge, eliminates all mechanically operated valves, except the air-starting valve, which has to be fitted to all types of Diesel engines. The scavenging air is admitted through ports on one side, sweeps round the cylinder and leaves through longer ports on the other side. On the return stroke of the piston, the inlet ports are covered before the exhaust ports, so that compression starts at about atmospheric pressure. In order to direct the air well up the cylinder, the piston head is usually provided with a lip as shown, and the cylinder head is sometimes shaped to suit.

A third method of scavenging has recently been introduced with the object of using the port scavenge and at the same time of starting compression at the pressure of the scavenging air, some 5 lb. per sq. in. above that of atmosphere. This method, known as double port scavenge, has two sets of inlet ports. The second set is controlled by a separate inlet valve which only opens after the exhaust ports have been covered by the returning piston. The proportions are so arranged that, when this second row of inlet ports are closed in their turn by the piston, the pressure inside the cylinder has risen to that of the scavenging pump.

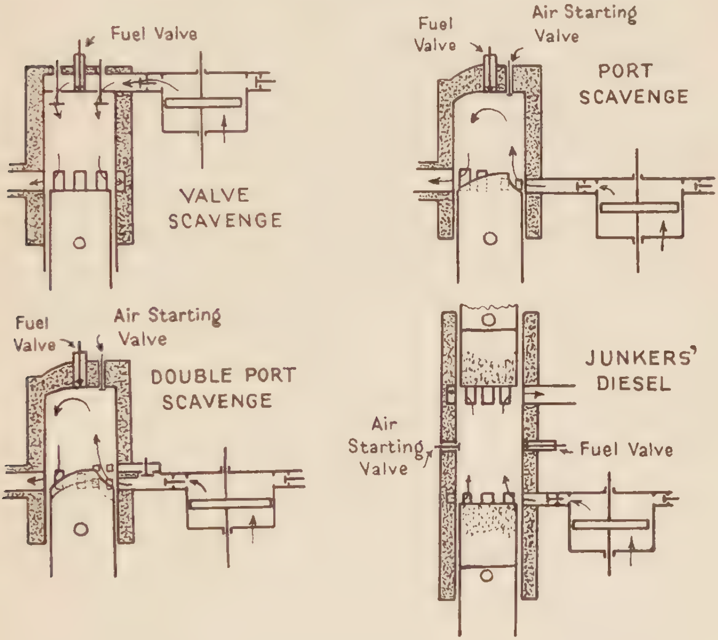

FIG. 20.—Methods of Scavenging Two-Stroke Cycle Diesel Engines.

A fourth method of scavenging two-stroke engines is that adopted by Junkers[1] from the Oechelhauser type of gas engine (Fig. 12). In this engine there are two pistons working in opposite directions and connected to the crankshaft by an arrangement similar to that in Fig. 12. Each piston uncovers a row of ports near the end of its outward stroke. The scavenging air enters at the lower end and sweeps straight through the cylinder to the exhaust ports at the top. In this type of engine, the fuel valves and air-starting valves are placed in the centre. Two fuel valves are often used on opposite sides of the cylinder and staggered about the width of the compression space. The spray from these fuel valves is flat and fan-shaped, and it is claimed with this arrangement that, as the opposed pistons move apart, they draw the compressed air after them and through the two flat oil sprays. In this way good intermingling is obtained. The compression ratio of these engines is very high and may be as much as 16 to 1.

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[1] *Zeitschrift des Vereines deutscher Ingenieure*, vol. 61 (1917), p. 283.

Junkers also introduced a method of cooling the pistons which may be referred to here. The portion inside each piston between the gudgeon pin and the piston face is enclosed and provided with a series of cooling fins (not shown) which project inwards. The space is half filled with oil which swings backwards and forwards as the pistons move up and down. It is claimed that in this way the oil extracts heat from the piston face and transfers it to the cooling fins, which in turn pass the heat on to the cooling jackets. Junkers called this system pendulum cooling.

Double-acting Diesel Engines.—Sooner or later attempts were sure to be made to combine the double-acting principle of the steam engine with the increased efficiency of the internal-combustion engine, but the great difficulties of cooling and the gas-tight joint which would be necessary round the piston rod, have hitherto been considered insurmountable obstacles. Although experimental units had been built,[1] the general tendency in design, and particularly in marine design, where power for weight is so important, has been to use two pistons, working in opposite directions, and to place the combustion space in the centre.

One or two continental firms,[2] however, as well as a well-known firm in Scotland,[3] have evolved double-acting Diesel engines which have a central piston and which work on the two-stroke principle. Such engines give a power indicator-card for each stroke, and one maker[4] estimates that a four-cylinder engine of this type will give the same power as twelve single-acting four-stroke cycle cylinders of the same diameter and stroke.

A diagram of the Scotch engine is shown in Fig. 21. It was designed by Mr. J. Maclagen and embodies other interesting features, besides the double-acting principle, in order to avoid cooling troubles and distortion of the lower combustion chamber owing to the presence of a piston rod.

In this engine, the piston has a central pin which comes through

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[1] For a general discussion of some of the difficulties see "Working Practice in the Design of Large Double-Acting Two-Stroke Engines," by A. E. L. Chorlton, Trans. N.E. Coast Inst. of Eng. and Shipbuilders, vol. 39 (1922), p. 59.

[2] For details of a continental double-acting Diesel, see "Die Nürnberger Grossöl-maschine," by W. Laudahn, *Zeit. des. Ver. deutsch. Ing.*, vol. 67 (1923), pp. 1093 and 1134. There is an English translation of this article in the American paper *Power*, vol. 59 (1924), pp. 603 and 642; see also p. 526.

[3] "Clyde Marine Oil Engines," A. L. Mellanby, Proc. Inst. Mech. Eng., June, 1923, p. 726.

[4] *Ibid.*, p. 728.

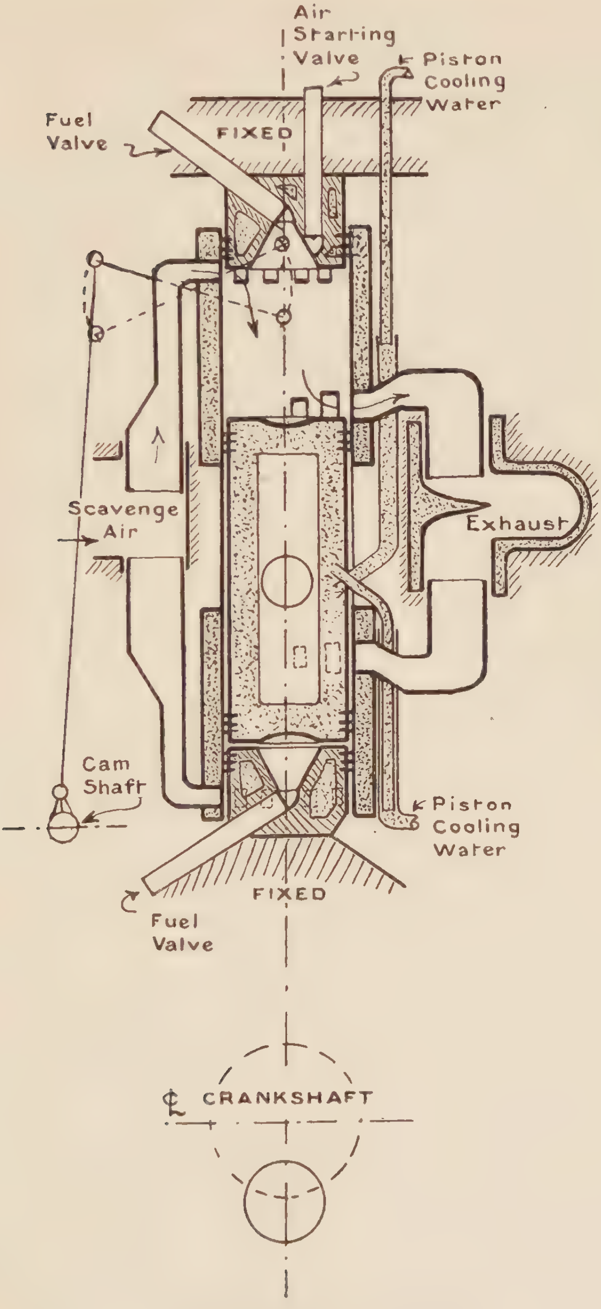

FIG. 21.—J. Maclagen's Double-Acting Diesel Engine.

a slot on each side of the cylinder and which is connected to the crankshaft by two rods working one on each side. This piston is very carefully water cooled by means of telescopic piping as shown. The cylinder is in two portions: a water-cooled liner which also oscillates up and down, and two fixed ends, which carry the fuel valves and which contain the two combustion spaces. The fixed portions are shown on the diagram as cross-hatched. Movement of the cylinder liner is obtained by means of a pair of rocking levers, one on each side, which work from the fuel valve cam-shaft. This movement is only about a third as great as that of the piston, but is sufficient to cover and uncover the ports for the scavenging air. The exhaust ports are uncovered by the piston in the usual way. It will be noted that telescopic joints are required for both the scavenging air inlet pipes and the two exhaust pipes. In order to maintain gas tightness, piston rings are fitted at each end of the piston, and also round each of the fixed combustion-chamber heads.

The Hot-bulb Crude Oil Engine.—About the year 1910, a new type of oil engine was introduced into England which has come to be known as the Hot-bulb, or Semi-Diesel oil engine. On account of its simplicity—there are no valves in the proper sense—it has achieved a great measure of popularity, particularly as launch or small cargo-boat engines, as also for auxiliary electric-lighting sets.

The principle on which this engine acts may be seen from Fig. 22. It works on a two-stroke or Clerk cycle, but obtains air for scavenging by means of what is called crank-case compression. This method of scavenging is different from those just outlined for two-stroke cycle Diesel engines. The combustion head is usually made partially spherical and the oil is injected through a fuel valve into this head, but away from the piston. The oil injection is mechanical, and no blast air is used. A small plunger pump is driven off the crankshaft and lifts a needle valve against a spring. In some makes this valve is actuated by a rocker lever as in Diesel-engine practice, in others it works on the method described in detail in the next section about solid-injection oil engines.

A portion of the combustion head or hot bulb (it is also sometimes called a vaporiser) is left unwater-jacketed, and when the engine is started up from cold, this part is brought to a dull red heat by means of a blow-lamp. After the engine gets well away this lamp can be dispensed with, and firing is automatic. The top of the piston is usually shaped as shown to guide the scavenging air round the inside of the cylinder. This air is drawn into an air-tight crank case through flap valves by means of the suction of the under side of the piston. It is then compressed to between 4 and 6 lb. per sq. in. above atmosphere by the next downward stroke of the piston and enters through the air ports as soon as they are uncovered by the piston top.

The variations in pressure in this air cycle may be seen from the indicator diagram on the left-hand side.

On the power side of the piston, air only is compressed on the upward stroke to about 180 lb. per sq. in. (abs.) and the oil is injected at or near the end of the compression stroke. Firing takes place at constant volume (though very occasionally there may be a trace of constant pressure as well) and the pressure rises to between 300 and 400 lb. per sq. in. (abs.).

Expansion follows on the outward stroke and the products of combustion commence exhaustion as soon as the exhaust ports (on the left of the cylinder) are uncovered by the piston.

The Solid-Injection Oil Engine.—The great improvements effected by the Diesel engine in the last twenty years or so have tended to obscure another development of the crude-oil engine which nevertheless has been going on for almost the same length of time. This development cannot be attributed, primarily, to a single inventor, but appears to have been evolved gradually by a number of manufacturers working more or less independently along similar lines. The great war gave considerable impetus to the development of this type of engine, but the methods of evolution which have been referred to in the first chapter, have resulted in the creation of a type of engine which is not always recognised as embodying a new principle. Some regard it as a modification of the hot-bulb engine from which they evolved it. Others claim it to be an off-shoot of the Diesel. It has not even got, at the present time, a distinctive name. It has

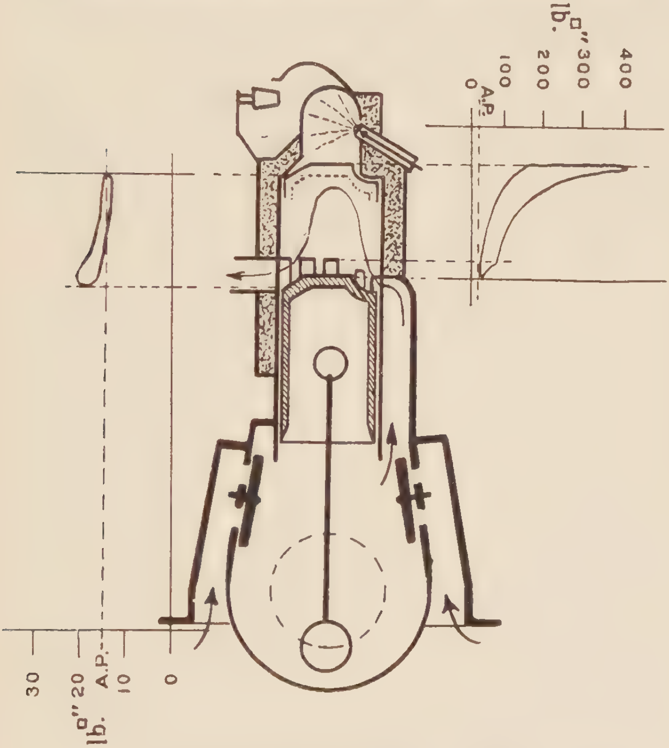

FIG. 22.—Hot-bulb Two-Stroke Oil Engine.

been called "cold-starting," to distinguish it from the hot-bulb engine which requires a preliminary heating up of a portion of the combustion chamber. On the other hand, it has been named "solid-injection," "airless-injection," or "mechanical-injection," in contradistinction to the Diesel engine which uses blast-air to inject the fuel. It is also known as "high-compression" to distinguish it from the lower or medium compressions of the "semi-Diesel" or of the hot-bulb types of crude-oil engines. Elsewhere the author has suggested that it might be called the "sprayer" oil engine, because the oil sprayer is the essential amd distinctive feature of the type. Of all these names, solid-injection seems at present to be the most common, and so it is used in this book.

The solid-injection engine proper has no hot bulb or vaporiser or other ignition device to fire the working substance, nor does it use an air-blast, like the Diesel, to introduce and to pulverise the fuel oil. It reverts back to the simplest form of internal-combustion engine, consisting of a working cylinder, and behind that a combustion space into which open an air-inlet valve, an air-starting valve, an oil sprayer, and, in the case of a four-stroke cycle engine, an exhaust valve.

The new principle is this, that ignition does not depend on any single factor, but on a variety of causes which have to be carefully balanced to give consistent and reliable results. The gas engine and the petrol engine use electric ignition. The Diesel engine compresses the air to such an extent before the fuel is introduced that the temperature of compression is more than ample to start combustion. The hot-bulb engine has a portion of the combustion space unjacketed so that there is always a "hot plate" which fires the mixture. The solid-injection engine relies partly on the compression of the air, partly on turbulence, but chiefly on the proper formation of the working substance. The ideal condition for ignition is a combustion chamber instantaneously filled at the end of compression with a homogeneous mixture of air and atomised oil in a high state of turbulence. The more nearly such an ideal is approached, the lower can the compression pressure be.

At the present time, each designer has his own combination for achieving such a result, and very few firms care to show exactly how it is done. The diagram in Fig. 23 does not, therefore, represent any particular make, but shows the principles on which the solid-injection engine works. The engine illustrated works on the four-stroke cycle, but solid injection can be used with the two-stroke cycle as well. The type of indicator card which may be obtained is shown at the top of the diagram. It is neither a constant-volume nor a Diesel-engine card, but a combination of both known as the dual-combustion cycle, or sometimes as the "mixed" cycle.

An elementary form of fuel pump is shown in the bottom left-hand corner. The plunger has a very short stroke and is cam operated from a lay shaft. By suitably proportioning the profile of this cam, the delivery of the oil to the fuel valve can be adjusted to time the injection so that the first portion of the mixture burns at constant volume whilst the remainder of the combustion is delayed and takes place at constant pressure. The significance of this card will be discussed in Chapter V. The oil is injected as shown into the middle of the sprayer, the spindle of which is made in two different diameters. The pressure of injection is great enough to overcome a stout spring in the upper part of the sprayer, with the result that the spindle is forced back and the oil penetrates into the combustion chamber through very fine holes in the nozzle. Occasionally a lighter oil may be used, either when starting or for helping a very thick crude oil. Provision is then made for this "pilot" oil to be introduced as shown. At the back of the sprayer there are two screw adjustments: one to regulate the lift of the needle valve and the other to adjust the tension of the spring. The pulverising of the fuel is done by the minuteness of the holes in a round-nosed nozzle. These holes, which may be only $\frac{1}{64}''$ in diameter, give an elongated spray rather like a fox's brush. There are many types of nozzles and arrangements used in practice which give different forms of spray, but in every case, to get the best mixture of atomised oil and air, the combustion chamber should be shaped to fit the spray combination and not the other way about. In the diagram there are supposed to be five holes of equal diameter in the nozzle, one in the

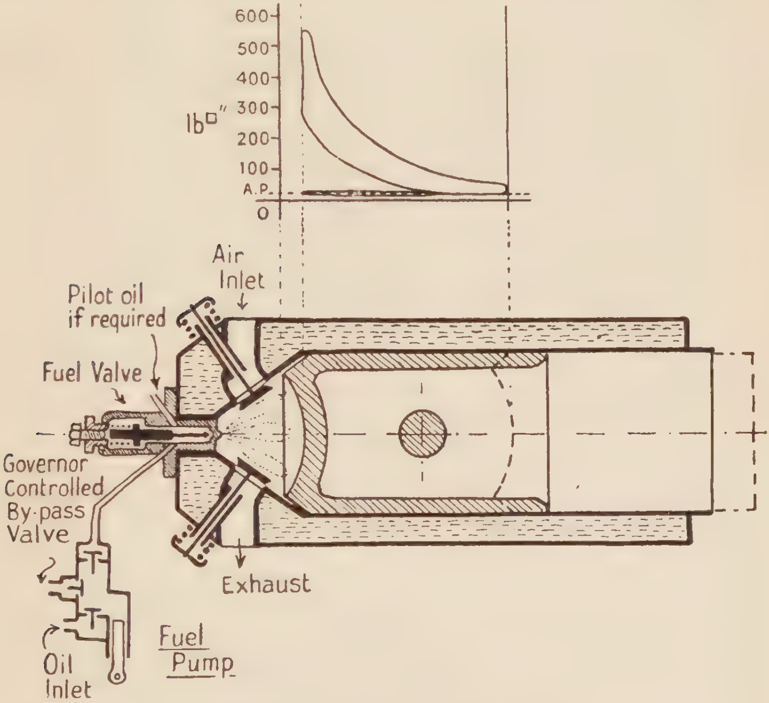

FIG. 23.—Solid-Injection Oil Engine.

centre and the other four grouped symmetrically about it, to give a cone angle of 70°. The combustion chamber is made cone shaped to suit this combination and the top of the piston is dished in spherically, so that, in theory, all the jets reach it together.

The air-inlet valve is on one side and the exhaust valve on the other. When the fuel is injected, both these valves should be as flush as possible with the conical face. An air-starting valve will be required at right angles to the plane of the diagram. The whole combustion head and the working cylinder are well water-jacketed, but the water inlet and outlet are not shown, as their position will depend on whether the cylinder is a horizontal or a vertical one. Either arrangement can be used with this type of engine.

It will be noted that the injection is directly timed by the strokes of the fuel pump plunger, the needle valve remaining lifted as long as the pressure is on. An oil bye-pass valve is therefore provided which is controlled by the governor in such a way that pressure is released earlier when the speed increases. The surplus, or overflow, oil passes back into the oil sump. This method is known as quantity governing. Results have already been obtained with the solid-injection engine which equal, and even exceed, those of the Diesel.

The Still Engine.—The principle of the engine invented by W. J. Still consists in utilising some form of internal combustion on one side of the working piston and steam on the other. The engine is primarily an internal-combustion engine, which utilises some of the heat that would otherwise be lost in the water jacket or in the exhaust, to generate steam. The steam so formed augments the steam generated in a small boiler alongside, and is then passed to the under side of the engine piston, where it does a certain amount of work. It may be recalled that the Simon engine of 1878 (p. 16) produced steam from the exhaust heat, but that, in this case, the steam was sprayed into the internal-combustion side of the piston and did not prove a success.

It is obvious that this principle can be applied to almost any type of internal-combustion engine, and the inventor himself has already experimented with petrol, oil, and gas.[1] In 1920, an experimental engine was built with opposed pistons on the Junkers principle. This engine worked with solid-injection, and had steam on the outer sides of the two pistons.[2] More recently still, the two-stroke, single-acting, solid-injection principle was used for the internal-combustion side, and a diagram of the thermal portion of this engine is shown in Fig. 24. Drawings and constructional details of the actual engine can be seen in *The Engineer*[3] for the latter half of 1923.

The working cylinder consists of a comparatively thin liner with a cone-shaped combustion chamber. This liner is ribbed on the outside with fins running lengthways and these fins are reinforced by an outer hoop which is designed to withstand the maximum working pressures inside the cylinder. In addition to providing a larger cooling surface, these fins or ribs also form convenient passages for controlling the flow of jacket water with which they are surrounded.

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[1] For details of these experiments, see F. D. Acland's paper before the Royal Society of Arts, May 26, 1919, abstracted in *The Engineer*, vol. cxxvii. (1919), p. 540.

[2] "Still and Sulzer Marine Eninges," by W. Denny; Inst. N.A., July 8, 1920. Reprinted in *Engineering*, vol. cx. (1920), p. 98.

[3] "The Scott-Still Marine Oil Engine," *The Engineer*, vol. cxxxvi. (1923), pp. 556 and 592. See also *The Engineer*, vol. cxxxiii. (1922), pp. 181 and 204.

The fuel-injection valve works on the same principle as the one just described in the solid-injection engine. The oil-fuel pump is not shown; it is driven off the main crankshaft of the engine. The engine works on the two-stroke cycle with port scavenge, and the piston top is suitably shaped for this purpose.

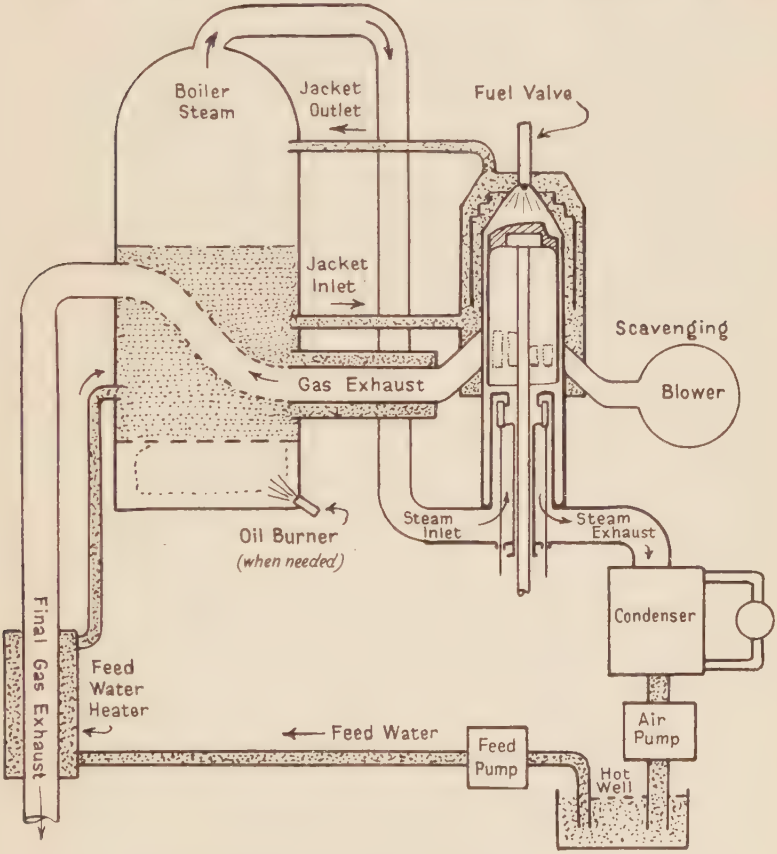

FIG. 24.—Two-Stroke Solid-Injection Still Engine of 1923.

The steam end of the cylinder is not jacketed and the inside of the trunk piston is used for the steam. The steam valves are placed well inside the cover formed by the piston. They are operated by an oil-relay system which gives easy control for reversing when necessary. The steam-inlet valve is shown on the left and the exhaust on the right.

The water circuit is as follows. Starting from the hot well (or its equivalent) in the bottom right-hand corner of the diagram, a feed pump passes it through a feed-water heater round the "final" exhaust pipe into the boiler. This boiler is heated, when running, by the exhaust products of the internal-combustion side of the engine, but oil-fuel burners are also fitted for starting up from cold or when otherwise required. The water is free to circulate round the exhaust pipe between the engine and the boiler, as well as round the jacket of the engine itself. In the jacket, the water is partially turned into steam and in so doing cools the cylinder walls. As the steam is generated at about 125 lb. per sq. in. (abs.) the temperature of this jacket remains at about 174° C. The steam and water return from the jacket into the upper portion of the boiler, where they are further heated by the exhaust gases, and the steam under pressure passes out at the top into the steam side of the piston. When more than one cylinder is used, it is possible to compound the steam side of the engine by using the remaining cylinders in parallel to form a low-pressure steam stage. Even triple expansion is possible. The underside of the first cylinder forms the H.P., the remaining cylinders the I.P., and an exhaust-steam turbine, to drive the scavenging blower, the L.P. Eventually the exhaust steam is condensed and returned to the hot well from whence it started. The mechanical efficiency of this engine is over 87 per cent., which is at least 10 per cent. higher than the Diesel engine and fuel consumptions of 0·36 lb. of oil per B.H.P. per hour have been obtained when running at full load.[1]

The Still engine is a good example of carefully considering a new idea in all its bearings and meeting each difficulty as it arose by experimental research.

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[1] Second Report of Marine Oil-Engine Trials Committee, Table 12, Proc. Inst. Mech. Eng., March, 1925.

CHAPTER IV

THE THERMODYNAMICAL RELATIONS OF A PERFECT GAS

Definition of a Perfect Gas.—A perfect gas is any substance, or mixture of substances, in a gaseous state which conforms with three simple relations. These relations may be briefly stated as follows :—

(*a*) The product of the pressure and the volume divided by the absolute temperature is constant for any state of a perfect gas.

(*b*) The internal energy of a given quantity of a perfect gas depends only on its (absolute) temperature.

(*c*) The ratio of the specific heat when measured at constant pressure to that at constant volume remains constant for all temperatures of a perfect gas.

None of these three relations is strictly true when applied to actual gases, but the deviations, in the case of the first two assumptions, are so slight that they may generally be neglected.

The third assumption, that the specific heats of gases do not vary with temperature, is not tenable when absolute solutions are required, but it may safely be used for comparative purposes.

There is another way of considering a perfect gas. All elementary substances, in theory, can be conceived as having three states : solid, liquid, and gaseous. In general, the particular state depends on the temperature of the substance. Those substances which are gaseous at normal temperatures and which require extreme cold to reduce them to the liquid state, behave approximately as perfect gases. In other words, substances which are stable gases under any temperature likely to be met with in practice (except in refrigerating machinery) have many physical properties in common and can, for convenience, be considered as behaving like perfect gases.

What these properties are should now be considered.

Boyle's Law (Mariotte's Law).—If the temperature of a gas remains unchanged whilst its volume is being altered by altering its pressure, then it is found that the volume varies inversely as the pressure.

Hence $$V \propto \frac{1}{P}$$

or $$PV = \text{a constant}$$

or $$P_1V_1 = P_2V_2$$

This relation holds whatever pressures and volumes are under consideration provided the temperature does not change. Hence, if 1 lb. of gas occupies v_1 cu. ft. (specific volume) at a pressure p_1 lb. per sq. in. it will occupy

$$\frac{p_1 v_1}{p_2} = v_2 \text{ cu. ft.} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

at a pressure of p_2 lb. per sq. in.

Conversely if 1 cu. ft. of gas at p_1 lb. per sq. in. weighs w_1 lb., at p_2 lb. per sq. in., 1 cu. ft. would weigh

$$w_1 \frac{p_2}{p_1} = w_2 \text{ lb.} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (2)$$

In other words, if the temperature of gas remains unchanged:

The specific volume varies inversely as the ratio of the pressures;

The specific weight varies directly as the ratio of the pressures.

Examples.—(1) 10 lb. of air (at 0° C.) are compressed isothermally (*i.e.* without change of temperature) from 14·0 lb. per sq. in. to 50 lb. per sq. in., what is the resulting volume?

1 lb. of air at 14·7 lb. per sq. in. and 0° C. occupies 12·387 cu. ft.

This is known as standard temperature and pressure, usually abbreviated to S.T.P.

Hence 10 lb. of air at 14·0 lb. per sq. in. occupies

$$\left(10 \times 12{\cdot}387 \times \frac{14{\cdot}7}{14{\cdot}0}\right) \text{ cu. ft.}$$

and 10 lb. of air compressed from 14·0 to 50·0 lb. per sq. in. occupy

$$10 \times 12{\cdot}387 \times \frac{14{\cdot}7}{14{\cdot}0} \times \frac{14{\cdot}0}{50{\cdot}0} = 36 \text{ cu. ft.}$$

(2) What is the specific wt. of air on a winter's day when the thermometer is at freezing point and the barometer drops to 29·0 inches of mercury?

Standard atmospheric pressure corresponding to 14·7 lb. per sq. in. is represented by 29·92 inches of mercury on the barometer.

Hence the sp. wt. of the air

$$= \frac{1}{12{\cdot}387} \times \frac{29}{29{\cdot}92}$$

$$= 0{\cdot}08073 \times \frac{29}{29{\cdot}92}$$

$$= 0{\cdot}07829 \text{ lb. per cu. ft.}$$

Charles's Law (Gay-Lussac's Law).—If a number of gases are heated under constant pressure (*e.g.* against the atmosphere) they will all expand an equal amount in volume for equal increases in temperature. This amount is found by experiment to be $\frac{1}{273}$ of their volume for each Centigrade degree rise in temperature.[1]

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[1] More correctly $\frac{1}{273{\cdot}1}$.

If a gas has a volume V_0 at 0° C. and is heated at constant pressure to t_1 °C., then its volume

$$V_1 = V_0 + V_0 \times \frac{t_1}{273}$$

$$= V_0\left(1 + \frac{t_1}{273}\right)$$

Similarly at any other temperature, t_2 °C.

$$V_2 = V_0\left(1 + \frac{t_2}{273}\right)$$

By division

$$\frac{V_2}{V_1} = \frac{\left(1 + \frac{t_2}{273}\right)}{\left(1 + \frac{t_1}{273}\right)} = \frac{t_2 + 273}{t_1 + 273}$$

Let $t + 273 = T$

then $\frac{V_2}{V_1} = \frac{T_2}{T_1}$

if P remains constant.

This may be written

$$\frac{w_1}{w_2} = \frac{T_2}{T_1} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (3)$$

if P remains constant.

Examples.—(1) A toy balloon holds 1 cu. ft. of gas at 15° C. What will be its volume if the room temperature increases to 20° C.?

$$\frac{V_2}{1} = \frac{20 + 273}{15 + 273}$$

$$V_2 = \frac{293}{288} = 1{\cdot}015 \text{ cu. ft.}$$

(2) The exhaust gases of an engine have a temperature of 275° C. as they come out into the atmosphere. What will be the reduction in their volume when they have cooled down to an atmospheric temperature of 15° C.?

$$\frac{V_2}{V_1} = \frac{15 + 273}{275 + 273} = \frac{288}{548} = 0{\cdot}52$$

or their volume will be reduced to about one-half.

Combination of the Laws of Boyle and of Charles.—By Boyle's Law the volume varies inversely as the pressure ratio, or

$$\frac{V_1}{V_2} = \frac{P_2}{P_1}$$

where T is constant.

Hence, if V_1 is the volume of a gas at an absolute temperature T_1 and a pressure P_1, then at some other pressure P_2 the new volume

$$V_2 = \frac{V_1 P_1}{P_2} \text{ at } T_1$$

By Charles's Law this volume varies directly as the temperature.

Hence, if T_1 changes to T_2 whilst P_2 remains constant, V_2 becomes

$$\frac{V_1 P_1}{P_2} \cdot \frac{T_2}{T_1}$$

or

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

This combination of the two laws of Boyle and of Charles is the characteristic equation of a perfect gas. It is usually written

$$\frac{PV}{T} = R \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where R is a constant quantity.

The Gas Constant.—If this relation is applied to unit weight of a gas, R is known as *the gas constant.*

Hence at S.T.P.

$$R = \frac{144 \times 14{\cdot}7 \times v}{273{\cdot}1} = 7{\cdot}75v \text{ ft.-lb. per lb. in °C.}$$

or

$$R = \frac{144 \times 14{\cdot}7 \times v}{491{\cdot}58} = 4{\cdot}306v \text{ ft.-lb. per lb. in °F.}$$

where v is the volume in cu. ft. of 1 lb of gas at S.T.P. If V is the volume of W lb. of gas at any pressure P and temperature T, then the gas constant

$$R = \frac{PV}{WT}$$

or

$$W = \frac{PV}{RT} \text{ lb.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

This is often a convenient relation to determine the weight of air or gas present in different parts of the cycles of internal-combustion engines.

For a mixture of gases at a temperature T and volume V

$$P_1 V = R_1 W_1 T, \quad P_2 V = R_2 W_2 T, \text{ etc.}$$

But by Dalton's Law of partial pressures

$$P_1 + P_2 = P$$

where P is the pressure of the resultant mixture.

Hence $$(P_1 + P_2)V = (R_1 W_1 + R_2 W_2)T = PV$$

and $$PV = RWT$$
$$= (R_1 W_1 + R_2 W_2)T$$

Hence the gas constant for the resultant mixture

$$R = \frac{R_1 W_1 + R_2 W_2}{W}$$

or R=the gas constants of each constituent gas multiplied by the percentage by weight of that gas in the mixture.

E.g. if air is assumed to consist of 23 per cent. by wt. of oxygen and 77 per cent. by wt. of nitrogen, and the gas constants for oxygen and nitrogen are taken as 86·86 and 99·24 respectively, then for air

$$R=\tfrac{23}{100}\times 86{\cdot}86+\tfrac{77}{100}\times 99{\cdot}24=96{\cdot}37 \text{ ft.-lb. per lb. in °C.}$$

A more accurate value is given in the table below.

Let w=wt. of a cu. ft. of gas at S.T.P., that is, at 0° C. and 14·70 lb. per sq. in.

Then from equation (5)

$$R=\frac{PV}{wT}=\frac{14{\cdot}7\times 144\times 1}{273{\cdot}1\times w}=\frac{7{\cdot}75}{w} \quad . \quad . \quad . \quad . \quad (6)$$

If δ=the density of a gas compared with air, then its molecular wt. $m=28{\cdot}95\delta$, if the molecular weight of air is taken as 28·95.

But 1 cu. ft. of air at S.T.P. weighs

$$\frac{1}{12{\cdot}387}=0{\cdot}08073 \text{ lb.}$$

Therefore
$$\delta=\frac{w}{0{\cdot}08073}$$

and
$$m=\frac{28{\cdot}95\,w}{0{\cdot}08073}=358{\cdot}6w \quad . \quad . \quad . \quad . \quad . \quad (7)$$

So that the gas constant can be expressed as a function of the molecular weight of a gas as follows:

$$R=\frac{7{\cdot}75\times 358{\cdot}6}{m}=\frac{2779}{m} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

in ft.-lb. per lb. per °C. units.

This may be converted to heat-units by dividing by J=1400.

Hence
$$R=\frac{1{\cdot}986}{m} \text{ C.H.U. per lb., per °C.} \quad . \quad . \quad (8a)$$

or gram. cal. per gram.

It may be noted here that the relation

$$mR=1{\cdot}986 \text{ gram. cal. per gramme-molecule}$$

is known as the universal gas constant.

TABLE OF PROPERTIES OF GASES AT S.T.P.

Gas.	Symbol.	w wt. of 1 cu. ft. in lb.	$v=\frac{1}{w}$ vol. of 1 lb. in cu. ft.	m molecular wt. (approx.)	$R=\frac{2779}{m}$ gas constant ft.-lb. per lb. in °C.
Air	·23O+·77N	0·08073	12·387	28·95	95·98
Oxygen	O_2	0·08944	11·18	32·0	86·86
Nitrogen . . .	N_2	0·07826	12·78	28·0	99·24
Hydrogen . . .	H_2	0·00559	178·9	2·0	13·90
Carbon monoxide	CO	0·07826	12·78	28·0	99·24
Carbon dioxide .	CO_2	0·1230	8·130	44·0	63·16

It may be noted that the expression

$$\begin{aligned} R &= \frac{2779}{m} \text{ ft.-lb. per lb. in °C.} \\ &= \frac{2 \times 1400}{m} \text{ very nearly} \\ &= \frac{2J}{m} \end{aligned}$$

So that the gas constant in ft.-lb. per lb. in °C may be determined from any of the following relations.

$$R = \frac{144pv}{T} = \frac{2J}{m} = \frac{7{\cdot}75}{w}$$

Examples.—Determine the gas constant for air, considered as a perfect gas, in

(*a*) English units and °C.;
(*b*) English units and °F.;
(*c*) Continental units;

given

1 lb. of dry air has a volume of 12·387 cu. ft. at 0° C. or 32° F. and 14·70 lb. per sq. in. (abs.).

1 kgm. of dry air has a volume of 0·77351 cu. metres at 0° C. and 1·03274 kilogrammes per sq. cm. (abs.).

The absolute zero for °C. $= 273{\cdot}1$ below 0° C.

The absolute zero for °F. $= 273{\cdot}1 \times \frac{9}{5} = 491{\cdot}58$ below 32° F.

Using the relation $\dfrac{PV}{T} = R$

(*a*) P is in lb. per sq. ft. $= 144 \times 14{\cdot}7$
V is in cu. ft. $= 12{\cdot}387$
T is in °C. absolute $= 0 + 273{\cdot}1$

$$\therefore \quad R = \frac{144 \times 14{\cdot}70 \times 12{\cdot}387}{273{\cdot}1} = 96{\cdot}01 \text{ ft.-lb. per lb., per °C.}$$

(*b*) P and V as before,
T is in °F. absolute $= 491{\cdot}58 = \frac{9}{5} \times 273{\cdot}1$

$$\therefore \quad R = 96{\cdot}01 \times \tfrac{5}{9} = 53{\cdot}33 \text{ ft.-lb. per lb. per °F.}$$

Alternatively $$R = \frac{144 \times 14{\cdot}7 \times 12{\cdot}387}{491{\cdot}58} = 53{\cdot}33$$

(*c*) P is in kgm. per sq. metre $= 1{\cdot}03274 \times 100^2$
V is in cu. metres $= 0{\cdot}77351$
T is in °C. absolute $= 0 + 273{\cdot}1$

$$R = \frac{1{\cdot}03274 \times 100^2 \times 0{\cdot}77351}{273{\cdot}1} = 29{\cdot}251$$

kilogramme metres per kilogramme per °C.

Note.—It is common practice to assign the following approximate values to the above units :

absolute zero for °C.=273 below 0° C.
absolute zero for °F.=492 below 32° F.

1 lb. of dry air has a volume of 12·39 cu. ft. at 14·7 lb. per sq. in. with these English values

R in Centigrade units =96·05
R in Fahrenheit units=53·26

In practice these figures are frequently taken as

R=96·00 ft.-lb. per lb. per °C.
=53·25 ft.-lb. per lb. per °F.

and for the Continental units

R=29·25 kgm.-metres per kgm. per °C.

(2) During a gas-engine trial the average pressure of a gas in a gas meter is p inches of water, and the barometer stands at b inches of mercury. If the average temperature of the gas is t° F. and the actual quantity of gas metered is V_1 cu. ft. per hr., deduce an expression to give the gas used per hr. at 29·921 in. of mercury and 32° F.

Density of mercury (at 60° F.)=13·56.

Here $P_1=b+\frac{p}{13\cdot56}$ in. of mercury $P_2=29\cdot921$ in. of mercury

V_1=vol. of gas at P_1 V_2=required answer

$T_1=t+460$ $T_2=32+460=492$

Hence
$$\frac{\left(b+\frac{p}{13\cdot56}\right)V_1}{t+460}=\frac{29\cdot921V_2}{492}=\frac{V_2}{16\cdot44}$$

$$V_2=\frac{16\cdot44\left(b+\frac{p}{13\cdot56}\right)V_1}{t+460}$$
$$=\frac{(16\cdot44b+1\cdot213p)V_1}{t+460}$$

Absolute Temperature and the Gas Thermometer.—It should be realised that the relation

$$\frac{PV}{WT}=R$$

which forms part of the definition of a perfect gas, may be used to construct a gas thermometer to determine values of T. Although in practice there are slight deviations, it is generally considered that if hydrogen is used for values between 0° C. and 100° C., nitrogen for values above 100° C., and helium for values below 0° C., the deviations between the temperature scales so determined and the above ideal relation are within the errors of practical experiment. In other words, it is possible to obtain readings on gas thermometers which are to all intents and purposes accurate enough to determine the absolute temperature T.

This can be seen from the following experimental results :[1]

	Ice-point.
Absolute hydrogen scale	273·03 ± 0·05 °C.
Absolute helium scale	273·11 ± 0·03 °C.
Absolute nitrogen scale	272·24 ± 0·05 °C.

In these experiments the volume remained constant and the pressure was varied from an initial value of 1000 mm. of mercury.

As far back as 1848, Lord Kelvin pointed out that it would be possible to define temperature without any reference to the properties of any particular substance such as a gas or liquid.[2] This definition is based on the supposition that an ideal engine taking in a quantity of heat Q_1 at a temperature T_1 and rejecting a lesser quantity of heat Q_2 at a lower temperature T_2 would show the following relation :—

$$\frac{T_1}{T_2}=\frac{Q_1}{Q_2}$$

In this engine the quantity of heat $Q_1 - Q_2$ would be converted into mechanical work, and as it is ideal there would be no other losses in the cycle. If a series of such engines are imagined, each one of which did an *equal amount of work* and passed on its surplus heat to the next engine, then a series of steps is obtained, in each of which there would be an equal drop in temperature. Such considerations follow regardless of the actual working substance which may be used in the engines, and the temperature scale would depend only on the amount of energy available in each engine.

Lord Kelvin based his reasoning on the Carnot cycle in which "The relation between motive power and heat is such that *quantities of heat*, and *intervals of temperature* are involved as the sole elements in the expression of mechanical effect to be obtained through the agency of heat."[3]

He states, "For every hundred units of heat converted into work by a perfect thermodynamic engine, 373·1 are taken from the source, and 273·1 rejected to the refrigerator, if the temperature of the source be that at which steam of water has a pressure of one atmosphere and the temperature of the refrigerator that at which ice melts."[4]

This method of reckoning temperature is known as the absolute thermodynamic scale, or the Kelvin thermodynamic scale, and the above figure of 273·1° C. at the ice-point is now considered to be correct to ±0·05°.

[1] See "The Realisation of Absolute Scale of Temperature," in "The Dictionary of Applied Physics" (1922), vol. i. p. 840. (Macmillan & Co.)

[2] "On an Absolute Thermometric Scale founded on Carnot's Theory of the Motive Power of Heat," "Collected Mathematical and Physical Papers" of Sir William Thomson (Lord Kelvin), vol. i., 1882, p. 100.

[3] *Ibid.* p. 102.

[4] Article on *Heat* by Sir William Thomson in the "Encyclopædia Britannica," 7th edition, 1880, vol. xi. p. 573.

For work, therefore, where all the data are accurate to four significant figures, the absolute zero of temperature should be taken as

$$-273{\cdot}1^\circ \text{ C.}$$

This figure corresponds to 491·58° F. below the freezing point 32° F., or

$$-459{\cdot}58^\circ \text{ F.}$$

In all other cases sufficient accuracy is obtained with the value

$$-273{\cdot}0^\circ \text{ C.}$$

or in Fahrenheit

$$-460^\circ \text{ F., or } 492^\circ \text{ F. below freezing point.}$$

Standard Temperature and Pressure (S.T.P.).—The characteristic equation of a perfect gas, $\frac{PV}{WT}=R$, is frequently used in internal-combustion engineering to reduce measurements of gaseous mixtures to a uniform basis for comparative, or other, purposes. The physical constants of a gas, such as its weight per cu. ft. or its volume per lb., are expressed in values which are always reduced to some standard temperature and pressure. The volume of a gas generated by a gas producer or used by a gas engine are similarly expressed.

This practice is so common that the resulting figures are frequently given without any indication that it is so, but sometimes the letters "at S.T.P." (at standard temperature and pressure) or "at N.T.P." (at normal temperature and pressure) are added.

In British engineering practice, V cu. ft. of a gas at S.T.P. means V cu. ft. at 0° C. (or 32° F.) and "standard" atmospheric pressure. The letters N.T.P. are very frequently taken to mean the same thing, though more logically they stand for V cu. ft. at the normal temperature of the air (say 15° C. or 60° F.) and normal atmospheric pressure, which is the same as standard atmospheric pressure.

This atmospheric pressure (A.P.) is usually taken as 14·70 lb. per sq. in., though if it is measured on a barometer a common British standard is 30 in. of mercury.

Both these measurements involve *weight*; the first directly and the second through the density of mercury, which in turn varies slightly with the temperature of the mercury.

At 0° C., the density of mercury is 13·5955 g. per c.c., whilst at normal temperature of 15° C. it is 13·5589 g. per c.c. Now 1 g. per c.c.$=$62·43 lb. per cu. ft.

$$=\frac{62{\cdot}43}{1728}=0{\cdot}03613 \text{ lb. per cu. in.}$$

Hence the density of mercury in British units is

$$13{\cdot}5955\times0{\cdot}03613=0{\cdot}4912 \text{ lb. per cu. in. at } 0^\circ \text{ C.}$$

or $$13{\cdot}5589\times0{\cdot}03613=0{\cdot}4898 \text{ lb. per cu. in. at } 15^\circ \text{ C.}$$

The barometric reading corresponding to 14·70 lb. per sq. in.

$$=\frac{14{\cdot}70}{0{\cdot}4912}=29{\cdot}92 \text{ in. of mercury at } 0° \text{ C.}$$

or

$$=\frac{14{\cdot}70}{0{\cdot}4898}=30{\cdot}01 \text{ in. of mercury at } 15° \text{ C.}$$

If, therefore, the barometer is read at a normal air temperature of approximately 15° C., or say 60° F., the British standard atmosphere of 14·70 lb. per sq. in. corresponds to 30·00 in. of mercury to 1 part in 3000.

The International standard temperature and pressure, which is used in accurate scientific research, is 0° C. and 760 mm. of mercury (at 0° C.) at latitude 45° and sea-level. The acceleration due to gravity at this latitude is 980·617 cm. per sec. per sec., though a conventional value of 980·665 C.G.S. units was fixed by an International Conference in 1901,[1] and is frequently used by American and continental engineers. In both cases, these figures represent 32·17 ft. per sec. per sec. to four significant figures. The conventional value for gravity in Britain is 32·20 ft. per sec. per sec., which is the true value at the latitude of Newcastle. This corresponds to 981·456 C.G.S. units, and consequently the mean standard atmosphere in Britain is

$$760\times\frac{980{\cdot}6}{981{\cdot}5}=759{\cdot}3 \text{ mm. of mercury at } 0° \text{ C. (lat. } 55°)$$

or 29·89 in. mercury at 0° C.

This corresponds to

$$29{\cdot}89\times 0{\cdot}4912=14{\cdot}696 \text{ lb. per sq. in.}$$

as compared with the usual British standard pressure of 14·70 lb. per sq. in.

The difference between the conventional British S.T.P. and the true international definition is only 1 part in 1500 if gravity is taken into account, and this is negligible unless the experimental data are within this accuracy; but the difference should be borne in mind when translating the valuable results which physicists are obtaining for internal-combustion engineers, or when comparing British with Continental or American practice.

To sum up:

The International and the British standard temperatures are both 0° C. or 32° F.

The International standard pressure to four significant figures

=760 mm. mercury at 0° C., lat. 45°, and sea-level;
=29·92 in. mercury at 0° C., lat. 45°, and sea-level;
=14·70 lb. per sq. in. at 0° C., lat. 45°, and sea-level.

The corresponding pressure to four significant figures in Britain when gravity is taken into account

=759·3 mm. mercury (at 0° C., lat. 55°, and sea-level);
=29·89 in. mercury (at 0° C., lat. 55°, and sea-level);
=14·69 lb. per sq. in. (at 0° C., lat. 55°, and sea-level).

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[1] "Troisième Conf. gen. des poids. et mes.," 1901, pp. 66-68.

The conventional standard pressure in Britain

=30 in. mercury at 15° C.
=29·92 in. mercury at 0° C.
=14·70 lb. per sq. in.

The International value of gravity is 32·17 ft. per sec. per sec. at lat. 45° sea-level.

The British value of gravity is 32·20 ft. per sec. per sec. at lat. 55° sea-level.

The letters N.T.P. (normal temperature and pressure) are better avoided unless they are definitely taken to mean 15° C. and 1 standard atmosphere.

So much work is published in Continental units that it is sometimes useful to know the conversion factors between the systems in use and British values.

On the Continent there are two methods of reckoning pressure: in "atmospheres" and in kgm. per sq. cm., and the two terms are often taken to mean the same thing.

The Continental "atmosphere" (at.) is the International standard of 760 mm. of mercury mentioned above and is taken at 0° C., where $g=980·617$ C.G.S. units. This corresponds to 14·696 lb. per sq. in. (to five significant figures) or 1·0333 kgm. per sq. cm. The kgm. per sq. cm. equals 0·9678 at. or 14·223 lb. per sq. in., which is 3·3 per cent. less than the "atmosphere." In considering any Continental data, however, it is better to ascertain from the context, if possible, whether the true atmosphere or the conventional one of 1 kgm. per sq. cm. is being used.

Joule's Law of Thermodynamics.—The second relation which may be said to define a perfect gas is (from p. 59):—

(*b*) The internal energy of a given quantity of a perfect gas depends only on its temperature.

The working substance (or gas) used in any heat engine may be considered as a heat reservoir, and the level of heat in that reservoir, or amount of heat available, represents the inherent energy which is present in that particular state of the substance. Just as water has a potential energy by virtue of its height, so can a gas be considered to have intrinsic or internal energy by virtue of the quantity of heat it contains. If, therefore, a gas changes its quantity or level of heat H_1 to another level H_2, without doing any external work, the measure of its change in internal energy E_1-E_2 can be represented in heat units as

$$E_1-E_2 \propto H_1-H_2$$

or

$$dE \propto dH$$

If this is considered per unit weight of gas

$$dE \propto dT$$

or

$$E=\int C_v dT$$

where C_v is the specific heat of the substance when measured under constant volume conditions (*i.e.* without doing external work).

In 1845, Dr. Joule published the results of some experiments which led him to conclude that "no change of temperature occurs when air is allowed to expand in such a manner as not to develop mechanical power."[1] This statement has been elaborated into Joule's Law of Thermodynamics, which says that "when a perfect gas is expanded without doing external work and without taking in or giving out heat, and therefore without changing its stock of internal energy, its temperature does not change."

In other words, the internal energy of a given quantity of a perfect gas depends only on the temperature and not on the pressure or the volume.

In general, when heat is taken in by unit weight of a perfect gas, the following relation holds good :—

Heat taken in=external work done+increase in internal energy :

$$dQ=\frac{dU}{J}+dE \text{ in heat units (lb. cal.)}$$

or

$$JdQ=dU+JdE \text{ in work units (ft.-lbs.)}$$

Applying Joule's thermodynamic law :

(*a*) *At constant volume.*

If unit weight of a perfect gas is heated at constant volume from an absolute temperature T_1 to T_2, no external work is done and $dU=0$.

Hence

$$\begin{aligned} JdQ&=JdE \\ &=K_v(T_2-T_1) \text{ in work units} \quad . \quad . \quad . \quad . \quad . \quad (1) \end{aligned}$$

on the assumption that the specific heat (K_v) is constant for the range of temperature (T_2-T_1).

(*b*) *At constant pressure.*

If, on the other hand, heat is taken in at a constant pressure of P lb. per sq. ft., and the volume of the gas changes from v_1 to v_2 cu. ft.

$$JdQ=K_p(T_2-T_1)$$

but the external work done at constant pressure

$$dU=P(v_2-v_1)$$

Hence the increase of internal energy under these conditions

$$JdE=K_p(T_2-T_1)-P(v_2-v_1)$$

But $Pv=RT$ for unit weight of a perfect gas.

Therefore

$$P(v_2-v_1)=R(T_2-T_1)$$

or

$$\begin{aligned} JdE&=K_p(T_2-T_1)-R(T_2-T_1) \\ &=(K_p-R)(T_2-T_1) \quad . \quad . \quad . \quad . \quad (2) \end{aligned}$$

But Joule's law says that the increase in internal energy depends only on the temperature.

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[1] "On the Changes of Temperature produced by the Rarefaction and Condensation of Air," by J. P. Joule. "Collected Scientific Papers" (1884), vol. i. p. 182. The actual apparatus used by Joule in these experiments can be seen on B corridor of the Manchester College of Technology.

Thus equations (1) and (2) are equal

or $$K_v(T_2-T_1)=(K_p-R)(T_2-T_1)$$

and the gas constant in work units

$$R=K_p-K_v \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

In this equation the specific heats are to be taken in work units per lb.

Thus $R=1400(C_p-C_v)$, where C_p and C_v are in lb.-cal. per lb., or, what is the same thing, gramme-cal. per gramme.

As the work unit is frequently more convenient for internal-combustion engineering calculations, K_p and K_v will, in general, be expressed in ft.-lb. per lb. in °C., in which case

$$R=K_p-K_v \text{ ft.-lb. per lb. in °C.} \quad . \quad . \quad . \quad (3a)$$

Dividing this relation by K_v

$$\frac{R}{K_v}=\frac{K_p}{K_v}-1$$

Let $$\frac{K_p}{K_v}=\gamma$$

then $$\frac{R}{K_v}=\gamma-1$$

or $$K_v=\frac{R}{\gamma-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Example.—What are the specific heats of air at S.T.P. when $\gamma=1{\cdot}4$?

$$R=96 \text{ ft.-lb. per lb. in °C.}$$

$$\therefore \ K_v=\frac{96}{1{\cdot}4-1}=240 \text{ ft.-lb. per lb. in °C.}$$

$$K_p=240\times1{\cdot}4=336 \text{ ft.-lb. per lb. in °C.}$$

[Note that $K_p-K_v=336-240=96=R$.]

More accurately $R=95{\cdot}98$ from Table p. 63

and $\gamma=1{\cdot}405$

Then $K_v=237$ ft.-lb. per lb. per °C.

and $K_p=333$ ft.-lb. per lb. per °C.

In heat units $C_v=\frac{237}{1400}=0{\cdot}169$ lb. cal. per lb. in °C.

and $C_p=\frac{333}{1400}=0{\cdot}238$ lb. cal. per lb. in °C.

Constant Specific Heats of a Perfect Gas.—The third definition suggested for a perfect gas on p. 59 is:—

(*c*) The ratio of the specific heat when measured at constant pressure to that at constant volume remains constant for all temperatures of a perfect gas.

No actual "permanent" gas obeys this relation, even if it approximates closely to the thermodynamic laws of Boyle, Charles, and Joule; nor is there anything in these laws from which it could be deduced.

For this reason, authorities like Sir Alfred Ewing[1] distinguish between the two cases by calling a gas which is supposed to conform to this third relation, as well as to the other two, an "ideal" gas, in contrast to the "perfect" gas which only obeys the first two relations.

The conception of a constant ratio of the two specific heats, that is of the numerical value of γ, is, however, of great convenience in a number of thermodynamical relations. It simplifies the resulting expressions and consequently the numerical calculations to such an extent that it is very commonly assumed to be an attribute of the perfect or ideal gas. As will be shown later, such an assumption is justified when relative or comparative results are required. Moreover, the deviations, in many cases, are either within the accuracy of experimental measurement or within the accuracy of other assumptions made in the calculation.

The Expansion or Compression of a Perfect Gas.—The thermodynamics of the internal-combustion engine are concerned with gases which are constantly changing their temperatures, pressures, and volumes, and the perfect gas is considered to do this in two main ways:

(*a*) Isothermal expansion or compression.

(*b*) Adiabatic expansion or compression.

(*a*) In the first case, by hypothesis, the temperature of the gas does not change, and the gas consequently obeys Boyle's law. For unit weight of gas

$$\mathrm{P}v = \text{constant}$$

The curve of this equation gives a rectangular hyperbola on the indicator diagram, and it is more frequently used in steam-engine practice than with internal-combustion engines.

(*b*) Adiabatic expansion or compression takes place without change of entropy, that is to say, without giving up or receiving any heat from an external source. It represents the way in which a perfect gas should behave in a gas-engine cylinder during the compression stroke or during the expansion stroke after combustion is complete.

Since from p. 70 for unit weight of a perfect gas

$$\mathrm{J}d\mathrm{Q} = d\mathrm{U} + \mathrm{J}d\mathrm{E} \text{ in work units (ft.-lb.)}$$

if no exchange of heat takes place during the change of condition

$$\mathrm{J}d\mathrm{Q} = 0$$

and

$$d\mathrm{U} = -\mathrm{J}d\mathrm{E}$$

That is to say, the external work done ($d\mathrm{U}$) is balanced by the change in internal energy ($\mathrm{J}d\mathrm{E}$).

But (in the limit)

$$\mathrm{J}d\mathrm{E} = \mathrm{K}_v d\mathrm{T}$$

and

$$d\mathrm{U} = \mathrm{P}dv$$

Hence

$$\mathrm{P}dv = -\mathrm{K}_v d\mathrm{T} \quad \ldots\ldots\ldots \quad (1)$$

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[1] See, for instance, the article on "Thermodynamics" in the "Dictionary of Applied Physics," vol. i. p. 928.

Again $$\mathrm{P}v=\mathrm{RT}$$
by differentiation
$$\mathrm{P}dv+vd\mathrm{P}=\mathrm{R}d\mathrm{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$
Dividing equation (2) by (1) to eliminate $d\mathrm{T}$.

Then $$\frac{\mathrm{P}dv+vd\mathrm{P}}{\mathrm{P}dv}=-\frac{\mathrm{R}}{\mathrm{K}_v}$$

or $$vd\mathrm{P}=-\mathrm{P}dv\left(1+\frac{\mathrm{R}}{\mathrm{K}_v}\right)$$

From equation (4), p. 71
$$\frac{\mathrm{R}}{\mathrm{K}_v}=\gamma-1,\ \text{hence } 1+\frac{\mathrm{R}}{\mathrm{K}_v}=\gamma$$

so that $$\frac{vd\mathrm{P}}{dv}=-\gamma\mathrm{P}$$

or $$\frac{d\mathrm{P}}{\mathrm{P}}+\gamma\frac{dv}{v}=0$$

By integration
$$\log_e \mathrm{P}+\gamma \log_e v=\text{a constant}$$
taking anti-logarithms
$$\mathrm{P}v^{\gamma}=\text{another constant}$$

This is an equation for an adiabatic curve of a perfect gas on the indicator diagram.

Again $$\mathrm{P}=\frac{\mathrm{RT}}{v}$$
multiply both sides by v^{γ}
$$\mathrm{P}v^{\gamma}=\mathrm{RT}v^{\gamma-1}=\text{a constant}$$
but R is a constant,

therefore $$\mathrm{T}v^{\gamma-1}=\text{another constant}$$

Thirdly $$v^{\gamma}=\frac{\mathrm{R}^{\gamma}\mathrm{T}^{\gamma}}{\mathrm{P}^{\gamma}}$$
multiply both sides by P
$$\mathrm{P}v^{\gamma}=\frac{\mathrm{R}^{\gamma}\mathrm{T}^{\gamma}}{\mathrm{P}^{\gamma-1}}=\text{a constant}$$
but R^{γ}=a constant,

therefore $$\frac{\mathrm{T}^{\gamma}}{\mathrm{P}^{\gamma-1}}=\text{another constant}$$

Collecting these results gives three adiabatic relations which combine any two of three variables of a perfect gas:
$$\mathrm{P}v^{\gamma}=\text{a constant}$$
$$\mathrm{T}v^{\gamma-1}=\text{a constant}$$
$$\frac{\mathrm{T}}{\underset{\mathrm{P}\ \gamma}{\gamma-1}}=\text{a constant}$$

If these relations are applied to the particular case of a perfect gas at P_1, T_1, and v_1 compressed in an internal-combustion engine to P_2, T_2, and v_2, where

$$\frac{v_1}{v_2} = \text{the compression ratio} = r$$

Then
$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = r^{\gamma}$$

$$\frac{v_2}{v_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}} = \frac{1}{r}$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{v_2}\right)^{\gamma-1} = r^{\gamma-1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

But since the specific volumes in these relations only appear as ratios, the same formulæ would hold good for $Wv = V$ cu. ft. of a perfect gas.

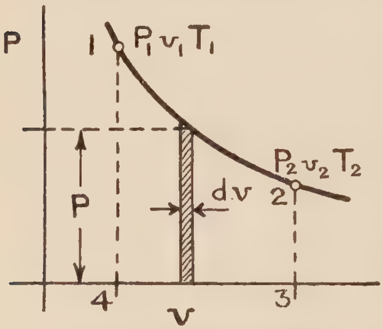

FIG. 25.—Expansion of a gas.

Again, if the laws of expansion and compression were according to the relation Pv^n = constant, where the index n has some other value than the ratio of the specific heats, the above proofs would still hold.

Work done by a Perfect Gas when changing its State.—Consider unit weight (1 lb.) of a perfect gas expanding from P_1, v_1, and T_1 to P_2, v_2, and T_2. The external work done by this expansion may be shown on the indicator diagram in Fig. 25 by the area, 1, 2, 3, 4, 1.

This area $U = \int_{v_1}^{v_2} P\,dv$

Here again there are two methods of expansion which may be considered.

(*a*) *Isothermal expansion* which follows the law Pv = a constant $= k$.

Hence
$$P = \frac{k}{v}$$

and
$$U = \int_{v_1}^{v_2} \frac{k}{v}\,dv = k\Big[\log_e v\Big]_{v_1}^{v_2} = k \log_e \frac{v_2}{v_1}$$

but
$$k = Pv$$

Hence
$$U = Pv \log_e \frac{v_2}{v_1} \qquad \text{. . . . . (1)}$$

If the process is reversed and the gas is compressed from 2 to 1, then the work is done on the gas.

If, further, v_2 is the cylinder volume before compression and v_1 the cylinder volume after compression, then $\frac{v_2}{v_1}=r$, the ratio of compression; and the work done on a gas when compressed isothermally

$$U=P_2v_2 \log_e r$$

This relation represents the ideal condition of an air compressor.

(*b*) *Adiabatic expansion.*—If the curve in Fig. 25 follows the law

$$Pv^\gamma=k \text{ and } P=\frac{k}{v^\gamma}$$

then, as before, the work done in expanding from 1 to 2

$$U=\int_{v_1}^{v_2}Pdv=\int_{v_1}^{v_2}\frac{k}{v^\gamma}dv$$

$$=k\left[\frac{1}{1-\gamma}\cdot v^{1-\gamma}\right]_{v_1}^{v_2}=\frac{k}{1-\gamma}(v_2^{1-\gamma}-v_1^{1-\gamma})$$

but $$k=P_1v_1^\gamma=P_2v_2^\gamma$$

therefore $$U=\frac{P_2v_2^\gamma v_2^{1-\gamma}-P_1v_1^\gamma v_1^{1-\gamma}}{1-\gamma}$$

$$U=\frac{P_2v_2-P_1v_1}{1-\gamma}=\frac{P_1v_1-P_2v_2}{\gamma-1} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Since $Pv=RT$ this may be written

$$U=\frac{R(T_1-T_2)}{\gamma-1}$$

or since $\frac{R}{\gamma-1}=K_v$ (equation (4), p. 71)

$$U=K_v(T_1-T_2) \text{ per lb. of gas}$$

If these expressions give a negative result, it follows that external work has been done *on* the gas. If the numerical answer is positive the work is done *by* the gas.

Rate of Reception or Rejection of Heat, assuming constant specific heats.—If the curve of expansion in Fig. 25 follows the law $Pv^n=k$ instead of $Pv^\gamma=k$, where n has any positive value greater than 1·0, the expansion need not then be adiabatic; but the same argument will hold for deriving the following expressions for the work done:

$$U=\frac{P_1v_1-P_2v_2}{n-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2a)$$

Let H equal the amount of heat received or rejected during the expansion (or compression), then

$$H=\text{work done}+\text{change in internal energy}$$

$$=U+K_v(T_2-T_1)$$

$$=\frac{P_1v_1-P_2v_2}{n-1}+\frac{R}{\gamma-1}\left(\frac{P_2v_2}{R}-\frac{P_1v_1}{R}\right)$$

$$=\frac{P_1v_1-P_2v_2}{n-1}-\frac{R}{\gamma-1}\left(\frac{P_1v_1}{R}-\frac{P_2v_2}{R}\right)$$

$$=\frac{P_1v_1-P_2v_2}{n-1}-\frac{P_1v_1-P_2v_2}{\gamma-1}$$

$$=(P_1v_1-P_2v_2)\left(\frac{1}{n-1}-\frac{1}{\gamma-1}\right)$$

$$=\frac{P_1v_1-P_2v_2}{n-1}\times\frac{\gamma-n}{\gamma-1}$$

$$=U\times\frac{\gamma-n}{\gamma-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

For small increments of volume,

$$U=Pdv$$

hence

$$dH=Pdv\times\frac{\gamma-n}{\gamma-1}$$

and the rate of heat reception or rejection per unit change in volume

$$\frac{dH}{dv}=P\times\frac{\gamma-n}{\gamma-1}$$

The value of n may be found approximately from the expansion (or compression) curve of an indicator diagram by measuring the actual values of P_1, P_2 and v_1, v_2 at two points on the curve.

Then, since

$$P_1v_1^n=P_2v_2^n=k$$

$$\log P_1+n\log v_1=\log P_2+n\log v_2$$

whence

$$n=\frac{\log P_2-\log P_1}{\log v_1-\log v_2}$$

Summary.—The conception of a perfect gas outlined in this chapter forms a convenient framework round which the thermal relations of the actual working substance can be built.

It has been shown that the *gas constant* (R) may be expressed in ft.-lb. per lb. of gas in °C. as follows:

$$R=\frac{PV}{WT}=\frac{144pv}{T}=\frac{7{\cdot}75}{w}=\frac{2J}{m}\text{ (very nearly) }\left[\text{actually }\frac{1{\cdot}986J}{m}\right]$$

R may also be expressed as a function of the specific heats as follows:

$$R=K_p-K_v=J(C_p-C_v)=K_v(\gamma-1)$$

For *isothermal change of volume* from a smaller volume V_1 to a larger volume V_2

$$PV=\text{a constant}$$

The work done

$$U=PV\log_e\frac{V_2}{V_1}$$

For adiabatic change of volume

$$PV^{\gamma}=\text{a constant}$$

or

$$TV^{\gamma-1}=\text{a constant}$$

or

$$\frac{T}{P^{\frac{\gamma-1}{\gamma}}}=\text{a constant}$$

The work done

$$U=\frac{P_1V_1-P_2V_2}{\gamma-1}$$

or

$$U=WK_v(T_1-T_2)$$

Further, if any quantity of a perfect gas at P_1, T_1, and V_1 is compressed to P_2, T_2, and V_2, where

$$\frac{V_1}{V_2}=\text{compression ratio}=r$$

Then

$$\frac{P_2}{P_1}=\left(\frac{V_1}{V_2}\right)^{\gamma}=r^{\gamma}$$

$$\frac{V_2}{V_1}=\left(\frac{P_1}{P_2}\right)^{\frac{1}{\gamma}}=\frac{1}{r}$$

$$\frac{T_2}{T_1}=\left(\frac{V_1}{V_2}\right)^{\gamma-1}=r^{\gamma-1}=\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$$

The standard temperature and pressure (S.T.P.) used in this book is

$p=14·70$ lb. per sq. in. abs.
$T=273°$ C.
or $t=0°$ C.

For air

the gas constant	$R=96$ ft.-lb. per lb. in °C. (S.T.P.)
and	$\gamma=1·4$
molecular weight	$m=28·95$
density	$\delta=1$
specific weight	$w=0·08073$ lb. per cu. ft. (S.T.P.)
specific volume	$v=12·387$ cu. ft. per lb. (S.T.P.)
specific heats	$K_v=237$ ft.-lb. per lb., per °C.
	$K_p=333$ ft.-lb. per lb., per °C.
	$C_v=0·169$ C.H.U per lb., per °C.
	$C_p=0·238$ C.H.U. per lb., per °C.

CHAPTER V

HEAT-ENGINE CYCLES OF OPERATION AND IDEAL EFFICIENCIES

Thermal Efficiency.—The thermal efficiency of any heat-engine cycle may be calculated from the relation :—

$$\frac{\text{Heat received less heat rejected}}{\text{Heat received}}$$

$$\eta = \frac{H - h}{H}$$

where H = the heat received (in heat units)
h = the heat rejected

This may be written

$$\eta = 1 - \frac{h}{H} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and in this form may be conveniently applied to find the thermal efficiency of different types of heat-engine cycles.

In the ideal internal-combustion engine, the heat is received whilst the fuel is undergoing combustion, and the heat is rejected whilst the exhaust valve is open. Strictly speaking this involves an alteration in the weight of the working substance, as the exhaust gases are in contact with the atmosphere ; but the assumption is made that the same weight of gases goes round and round the cycle in a closed circuit with the further assumption that only the working strokes in the cylinder are taken into consideration. Two-stroke engines require some form of pump to scavenge the cylinder and, in some cases, to introduce the charge, whilst engines working on the four-stroke principle have two pumping strokes of the main piston during which no useful work is done. The work absorbed by either of these two actions is not generally taken into account when considering ideal efficiencies.

The further assumption will be made, for the present, that the specific heats of the working substance do not vary with increasing temperatures.

The ideal engine also works adiabatically and in most cases postulates complete combustion, though, in practice, there may be heat exchanges between the working medium and the container walls, or after burning may take place during the period of expansion.

All these various causes modify the efficiency which could be obtained in this way, and the ratio

$$\frac{\text{actual thermal efficiency}}{\text{ideal thermal efficiency}}$$

is known as the relative efficiency, the efficiency ratio, or sometimes as the coefficient of performance.

In this chapter will be considered the various types of heat-engine cycles of operation which are (or have been) found in practice and their ideal efficiencies. The heat received (H) and the heat rejected (*h*) can be determined from diagrams of the cycles of operation, which may be plotted in different ways.

The most common form is the indicator card on which absolute pressures are plotted as ordinates and the total volume behind the piston as abscissæ. But a temperature-volume diagram, in which temperature replaces pressure, is equally suitable, and the most scientific method of all is to show the heat account on a temperature-entropy diagram, where entropy is plotted on the abscissæ scale instead of volume.

Of these three methods, the first is the most practical because it is possible either to obtain actual diagrams from specific engines with which to compare the ideal diagram, or to construct diagrams which approximate very closely to what should pertain in an actual engine.

The Lenoir Engine of 1860 (p. 10).—The ideal indicator card for this type of engine, which is now quite obsolete, is shown in Fig. 26. The suction stroke, assuming no cylinder clearance, took place from 1 to 2, and explosion followed at constant volume, *i.e.* heat was received from 2 to 3. The products of combustion expanded adiabatically from 3 to 5 whilst doing useful work on the piston, and exhaust took place from 5 to 1, during which period heat was rejected. The useful work done is shown by the area of the indicator card 2, 3, 5, 2, so that, as far as the cycle of operation is concerned, heat was rejected from 5 to 2.

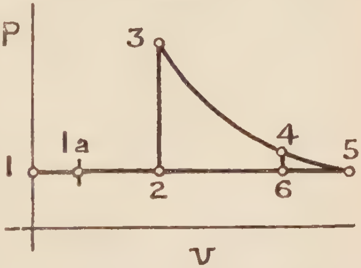

FIG. 26.—Ideal Diagram for Lenoir Engine.

In an actual engine, there was clearance at both ends, so that the above operations took place as follows :—

Suction	1*a* to 2
Explosion	2 to 3
Expansion	3 to 4
Exhaust	4 to 6 to 1*a*

In this case, the area of the indicator card is represented by 2, 3, 4, 6, 2, and the toe of the diagram is cut off.

Hence heat was rejected from 4 to 6 and also from 6 to 2.

Consider first the efficiency (per lb. of working substance) when no clearance is assumed.

Heat received $H=C_v(T_3-T_2)$, the specific heat at constant volume being used because the explosion took place at constant volume.

Heat rejected $h=C_p(T_5-T_2)$
In this case heat is rejected at constant pressure, so C_p is used.

Thermal efficiency

$$\eta=1-\frac{h}{H}=1-\frac{C_p(T_5-T_2)}{C^v(T_3-T_2)}$$

but $$\frac{C_p}{C_v}=\gamma$$

therefore $$\eta=1-\gamma\frac{(T_5-T_2)}{(T_3-T_2)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Equation 2 is all that is really required to determine the ideal efficiency of the cycle, but it may be simplified for convenience in calculation as follows:

Let $$\frac{P_3}{P_2}=\alpha \text{ and } \frac{V_5}{V_2}=\beta$$

then T_2=approximately the suction temperature in degrees absolute.

$$T_3=\alpha T_2$$

$$T_5=\left(\frac{P_5}{P_3}\right)^{\frac{\gamma-1}{\gamma}} T_3 \quad =\alpha\left(\frac{1}{\alpha}\right)^{\frac{\gamma-1}{\gamma}} T_2 \quad =\alpha^{\frac{1}{\gamma}} T_2$$

Substituting these values in equation (2)

$$\eta=1-\gamma\frac{\alpha^{\frac{1}{\gamma}}T_2-T_2}{\alpha T_2-T_2}$$

$$=1-\gamma\frac{\alpha^{\frac{1}{\gamma}}-1}{\alpha-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2a)$$

alternatively $$T_5=\beta T_2$$

whence $$\eta=1-\gamma\frac{\beta-1}{\alpha-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2b)$$

If clearance is taken into account, then the heat received

$$H=C_v(T_3-T_2) \text{ as before}$$

but the heat is rejected both at constant volume and at constant pressure, or

$$h=C_v(T_4-T_6)+C_p(T_6-T_2)$$

the thermal efficiency

$$\eta=1-\frac{C_v(T_4-T_6)+C_p(T_6-T_2)}{C_v(T_3-T_2)}$$

$$=1-\frac{(T_4-T_6)+\gamma(T_6-T_2)}{(T_3-T_2)} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

As before let $\frac{P_3}{P_2}=\alpha$

and also $\frac{V_4}{V_2}=\beta$

then $T_2=$suction temperature

$T_3=\alpha T_2$ as before

$T_4=\left(\frac{1}{\beta}\right)^{\gamma-1}T_3 \quad =\alpha\left(\frac{1}{\beta}\right)^{\gamma-1}T_2$

$T_6=\beta T_2$

Substituting in equation (3) and eliminating T_2

$$\eta=1-\frac{\left\{\alpha\left(\frac{1}{\beta}\right)^{\gamma-1}-\beta\right\}+\gamma(\beta-1)}{(\alpha-1)} \quad . \quad . \quad . \quad (3a)$$

$$=1-\frac{(\alpha\beta^{1-\gamma}-\beta)+\gamma(\beta-1)}{(\alpha-1)}$$

$$=1-\frac{\beta(\alpha\beta^{-\gamma}-1)+\gamma(\beta-1)}{\alpha-1} \quad . \quad . \quad . \quad . \quad (3b)$$

When there is no clearance $T_4=T_6$

or $$\alpha\left(\frac{1}{\beta}\right)^{\gamma-1}=\beta$$

equation (3*a*) becomes equation (2).

Example.—Consider the card shown in Fig. 2. The maximum pressure would have to be estimated from the curved peak, say 50 lb. per sq. in. (abs.).

Then $p_3=50$ lb. per sq. in. (abs.)

$p_2=14{\cdot}7$ lb. per sq. in. (abs.)

$\alpha=\frac{50}{14{\cdot}7}=3{\cdot}4$

$V_5=100$ per cent.

$V_2=42$ per cent. (approx.)

$\beta=\frac{100}{42}=2{\cdot}38$

$\gamma=1{\cdot}4$ (for air)

Then, by equation (2*a*)

$$\eta=1-1{\cdot}4\,\frac{3{\cdot}4^{\frac{1}{1\cdot4}}-1}{3{\cdot}4-1}$$

$$=1-1{\cdot}4\,\frac{2{\cdot}39-1}{3{\cdot}4-1}$$

$$=1-0{\cdot}81$$

$$=19{\cdot}0 \text{ per cent.}$$

Alternatively, by equation (2*b*)

$$\eta = 1 - 1{\cdot}4\frac{2{\cdot}38 - 1}{3{\cdot}4 - 1}$$

$$= 19{\cdot}5 \text{ per cent.}$$

If clearance is taken into account, then

$$\beta = \text{about } 2$$

and the efficiency, by equation (3*b*)

$$\eta = 1 - \frac{2(3{\cdot}4 \times 2^{-1{\cdot}4} - 1) + 1{\cdot}4(2 - 1)}{3{\cdot}4 - 1}$$

$$= 1 - \frac{2(3{\cdot}4 \times 0{\cdot}38 - 1) + 1{\cdot}4}{2{\cdot}4}$$

$$= 1 - \frac{1{\cdot}98}{2{\cdot}4}$$

$$= 1 - 0{\cdot}825$$

$$= 17{\cdot}5 \text{ per cent.}$$

or 1·5 per cent. less than the case without clearance.

It is interesting to note that the exhaust pressure p_4 may be calculated from the relation

$$\frac{p_3}{p_4} = \left(\frac{V_4}{V_3}\right)^{\gamma} = \beta^{\gamma}$$

$$= 2^{1{\cdot}4} = 2{\cdot}638$$

Hence $$p_4 = \frac{50}{2{\cdot}638} = 19 \text{ lb. per sq. in. (abs.)}$$

The Otto and Langen Free Piston Engine (p. 12, Chap. II.).—The theoretical diagram which is usually given for this engine may be seen in Fig. 27. It is assumed that air and gas are drawn in at atmospheric pressure from 1 to 2, that explosion takes place at constant volume from 2 to 3, followed by adiabatic expansion from 3 to 4 whilst the rising piston is gradually brought to rest; and lastly it is assumed that isothermal compression takes place from 4 to 2 during the working stroke of the engine. The exhaust (theoretically) taking place at constant pressure from 2 to 1, outside the cycle of operations.

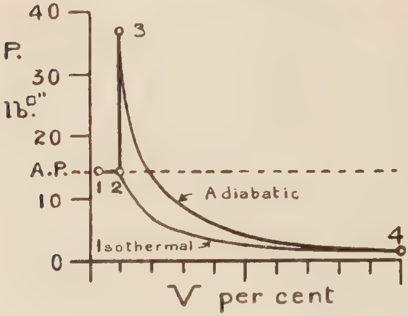

FIG. 27.

On this hypothesis it has been customary to work out the ideal efficiency as follows:

Heat received at constant volume between 2 and 3 $= K_v(T_3 - T_2)$ work units. As the remainder of this stroke from 3 to 4 is assumed to be adiabatic, no further heat is gained (or lost).

Heat rejected at isothermal compression between 4 and 2 $= P_4V_4 \log_e \left(\frac{V_4}{V_2}\right)$ (by equation (1), p. 74).

As this stroke is assumed to take place at constant temperature, there is no change in the internal energy.

But $$P_4V_4 = RT_4 \text{ or } RT_2$$

and $$\frac{V_4}{V_2} = r$$

$$\therefore \eta = 1 - \frac{RT_2 \log_e r}{K_v(T_3 - T_2)} \quad . \quad . \quad . \quad . \quad . \quad (4a)$$

again $$T_3 = \alpha T_2 \text{ and } R = K_p - K_v$$

$$\therefore \eta = 1 - \frac{(\gamma - 1) \log_e r}{\alpha - 1} \quad . \quad . \quad . \quad . \quad . \quad (4b)$$

Lastly $$\alpha = \frac{T_3}{T_2} = \frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = r^{\gamma-1}$$

so that $$\log_e \alpha = (\gamma - 1) \log_e r$$

and $$\eta = 1 - \frac{\log_e \alpha}{\alpha - 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4c)$$

A common value for the compression ratio in this type of engine was about 10, and if the card outlined above is drawn to scale as in Fig. 27, starting from $V_2 = 10$ per cent. and $P_2 = 14{\cdot}7$ lb. per sq. in. (abs.), it will be found that $P_3 = 36{\cdot}9$ lb. per sq. in. (abs.) and that $P_4 = 1{\cdot}47$ lb. per sq. in. (abs.).

The efficiency would therefore be

$$\eta = 1 - \frac{\log_e \frac{36{\cdot}9}{14{\cdot}7}}{\left(\frac{36{\cdot}9}{14{\cdot}7}\right) - 1}$$

$$= 39 \text{ per cent.}$$

The pressure p_4 never reached anything like 1·47 lb. per sq. in. (abs.) and was usually in the neighbourhood of 5 lb. per sq. in. (abs.). The theoretical card on the above hypothesis would then look like Fig. 28. An actual card as shown in Fig. 4, p. 15, is added in dotted lines.

There is so much after burning that combustion is hardly complete before the end of the upward stroke.

It would appear to be more in accordance with the facts to say that expansion is isothermal instead of adiabatic and that compression follows some law such as PV^n=const., where n is less than unity.

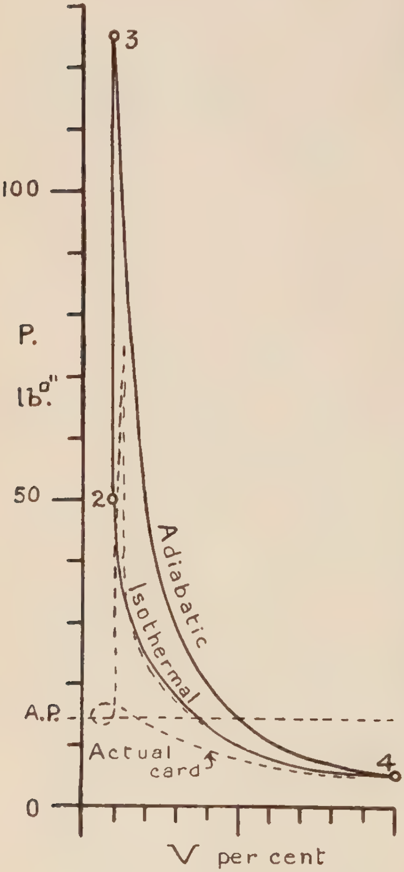

FIG. 28.

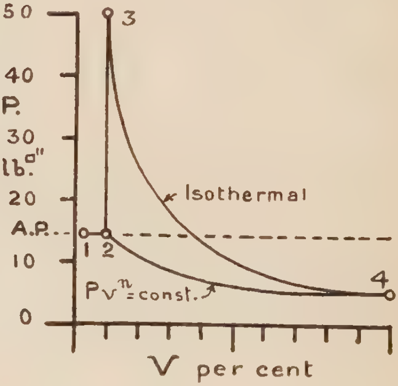

FIG. 29.

The ideal card would then look like Fig. 29. An approximate value for the index n is obtained thus :—

$$P_4V_4{}^n = P_2V_2{}^n$$

$$\log 5 + n \log 100 = \log 14{\cdot}7 + n \log 10$$

$$0{\cdot}699 + 2n = 1{\cdot}1673 + n$$

$$n = 1{\cdot}1673 - 0{\cdot}699$$

$$= 0{\cdot}468$$

$$n = \tfrac{1}{2} \text{ very nearly.}$$

The ideal efficiency should then be deduced as follows :

Heat received at constant volume

$$\text{from 2 to 3} = K_v(T_3 - T_2)$$

and during isothermal expansion

$$\text{from 3 to 4} = P_3V_3 \log_e\left(\frac{V_4}{V_3}\right)$$

Heat rejected according to PV^n = const.

$$= \frac{P_4V_4 - P_2V_2}{1 - n} + K_v(T_4 - T_2)$$

but $K_v(T_4 - T_2) = K_v(T_3 - T_2)$ by hypothesis

$$= \frac{R}{\gamma - 1}\left(\frac{P_4V_4}{R} - \frac{P_2V_2}{R}\right)$$

$$= \frac{P_4V_4 - P_2V_2}{\gamma - 1}$$

$$\therefore\ \eta = 1 - \frac{\dfrac{P_4V_4 - P_2V_2}{1-n} + \dfrac{P_4V_4 - P_2V_2}{\gamma - 1}}{P_4V_4 \log_e r + \dfrac{P_4V_4 - P_2V_2}{\gamma - 1}} \quad . \quad . \quad . \quad (5)$$

Example.—The ideal card in Fig. 29 which corresponds to the actual card of Fig. 28 has the following values:

$P_4 = 5$ lb. per sq. in. (abs.)
$P_2 = 14{\cdot}7$ lb. per sq. in. (abs.)
$P_3 = 50$ lb. per sq. in. (abs.)

The conversion in this expression to lb. per sq. ft. is unnecessary, as P occurs in every term.

$V_4 = 100$ per cent.
$V_2 = 10$ per cent.
$V_3 = 10$ per cent.
hence $r = 10$ and $\log_e r = 2{\cdot}3$
again $\gamma = 1{\cdot}4$ and
$n = 0{\cdot}5$ (approximately)

$$\therefore\ \eta_2 = 1 - \frac{\dfrac{500 - 147}{0{\cdot}5} + \dfrac{500 - 147}{0{\cdot}4}}{500 \times 2{\cdot}3 + \dfrac{500 - 147}{0{\cdot}4}}$$

$$= 1 - \frac{706 + 882}{1150 + 882}$$

$$= 1 - 0{\cdot}78 \text{ or } 22 \text{ per cent.}$$

as compared to 39 per cent. on the usual theory. A stricter analysis still would tend to show that the ideal card is isothermal between maximum pressure (P_3) and about the atmospheric line and adiabatic onward to P_4. This would still further reduce the ideal thermal efficiency, whilst at the same time slightly decreasing the value of r.

The Brayton Engine of 1872 and the Simon adoption of this Engine in 1878.—Sir Dugald Clerk and Burls show[1] that these two engines worked on what is now known as the Joule or true constant-pressure cycle.

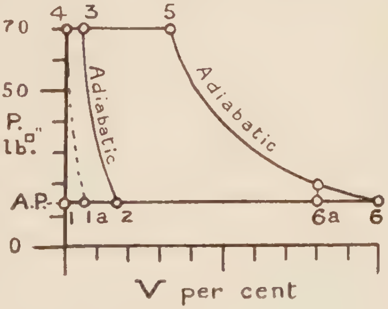

FIG. 30.

There were two cylinders, a pumping cylinder and a working cylinder. The pump (neglecting clearance) drew in a mixture of oil vapour and air from 1 to 2, and compressed it adiabatically from 2 to 3. The connection between the two cylinders was then opened and the charge was transferred to the working cylinder, at constant pressure, from 3 to 4. The ideal indicator diagram for the pump when clearance was neglected, was therefore 1, 2, 3, 4. The effect of clearance in the pump would be to cause the line 4 to 1 to be a second adiabatic, 4 to 1*a* as shown in dotted lines. In the working cylinder the charge burnt at constant pressure from 4 to 5, and then expanded adiabatically down to atmospheric pressure, 5 to 6.

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[1] "Dictionary of Applied Physics," vol. i. p. 280 (1922).

The exhaust stroke (neglecting clearance) took place from 6 to 1.

The nett work of the ideal cycle is therefore shown by the card 2, 3, 5, 6, 2.

In practice exhaust took place before atmospheric pressure was reached, and the toe of the diagram was cut off at 6*a*.

Consider the ideal card:

Heat received at constant pressure from 3 to 5 $=C_p(T_5-T_3)$

Heat rejected at constant pressure from 6 to 2 $=C_p(T_6-T_2)$

$$\eta=1-\frac{C_p(T_6-T_2)}{C_p(T_5-T_3)} \quad . \ . \ . \ . \ . \ . \ . \quad (6a)$$

This expression may be simplified as follows:

Since $$\frac{PV}{T}=R$$

therefore $$\frac{P_5V_5}{T_5}\times\frac{T_3}{P_3V_3}=\frac{P_6V_6}{T_6}\times\frac{T_2}{P_2V_2}$$

But both expansion and compression curves are adiabatics, therefore

$$\frac{V_5}{V_3}=\frac{V_6}{V_2}$$

Also, by hypothesis, $P_5=P_3$ and $P_6=P_2$

$$\therefore \ \frac{T_3}{T_5}=\frac{T_2}{T_6} \quad \text{or} \quad \frac{T_6}{T_5}=\frac{T_2}{T_3}$$

Hence $$\frac{(T_6-T_2)}{(T_5-T_3)}=\frac{T_6}{T_5} \quad \text{or} \quad \frac{T_2}{T_3}$$

and $$\eta=1-\frac{T_6}{T_5} \quad \text{or} \quad 1-\frac{T_2}{T_3} \quad . \ . \ . \ . \ . \ . \quad (6b)$$

but $$\frac{T_2}{T_3}=\left(\frac{V_3}{V_2}\right)^{\gamma-1}=\left(\frac{1}{r}\right)^{\gamma-1} \quad \text{(p. 77)}$$

$$\therefore \ \eta=1-\left(\frac{1}{r}\right)^{\gamma-1} \quad . \ . \ . \ . \ . \ . \ . \ . \ . \quad (6c)$$

The Otto "Silent" Gas Engine of 1876.—The cycle of this engine, seen in Fig. 31, has become the standard ideal cycle for internal-combustion engines and is known as the Otto cycle or the true constant-volume cycle. The working substance is drawn in at atmospheric pressure from 1 to 2, adiabatically compressed between 2 and 3, exploded at constant volume from 3 to 4, expanded adiabatically from 4 to 5, and exhausted first at constant volume from 5 to 2 and then at atmospheric pressure between 2 and 1.

The suction and portion of exhaust stroke between 1 and 2 are not taken into account in the ideal case, and a definite amount of the working fluid is considered to repeat the closed cycle, 2, 3, 4, 5, 2. The efficiency may be derived as follows :—

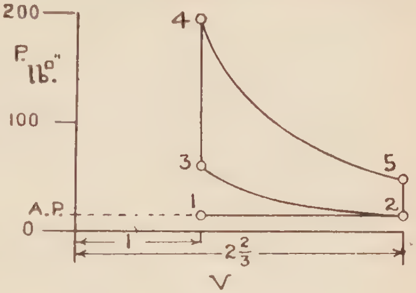

FIG. 31.

Heat received at constant volume from 3 to $4 = C_v(T_4 - T_3)$

Heat rejected at constant volume from 5 to $2 = C_v(T_5 - T_2)$

efficiency
$$\eta = 1 - \frac{C_v(T_5 - T_2)}{C_v(T_4 - T_3)} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (7a)$$

Applying a similar analysis to this expression as in the previous case of the constant-pressure cycle

$$\frac{P_5V_5}{T_5} \times \frac{T_2}{P_2V_2} = \frac{P_4V_4}{T_4} \times \frac{T_3}{P_3V_3}$$

but
$$V_5 = V_2 \text{ and } V_4 = V_3$$

and, since both the curves are adiabatics,

$$\frac{P_5}{P_2} = \frac{P_4}{P_3}$$

Hence
$$\frac{T_2}{T_5} = \frac{T_3}{T_4} \text{ or } \frac{T_2}{T_3} = \frac{T_5}{T_4}$$

therefore
$$\frac{(T_5 - T_2)}{(T_4 - T_3)} = \text{either } \frac{T_2}{T_3} \text{ or } \frac{T_5}{T_4}$$

and
$$\eta = 1 - \frac{T_2}{T_3} \text{ or } 1 - \frac{T_5}{T_4} \quad . \; . \; . \; . \; . \; . \quad (7b)$$

but
$$\frac{T_2}{T_3} = \left(\frac{V_3}{V_2}\right)^{\gamma-1} = \left(\frac{1}{r}\right)^{\gamma-1} \text{ p. 77}$$

therefore
$$\eta = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (7c)$$

which is the same expression as that for the true constant-pressure efficiency. A committee of the Institution of Civil Engineers,[1] which first developed this expression, proposed that it should be used as a standard of efficiency for all internal-combustion engines with an ideal working substance which had the properties of air. The value of γ for air may be taken as 1·4, whence

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[1] Preliminary Report, Proc. Inst. C.E., vol. clxii. (1905), pp. 307–338.

$$\text{The } \textit{air standard efficiency} = 1 - \left(\frac{1}{r}\right)^{0.4}$$

where r is the compression ratio as usual.

The only variable in the air standard efficiency is the compression ratio r. Fig. 32 shows the curve of this expression up to $r=16$.

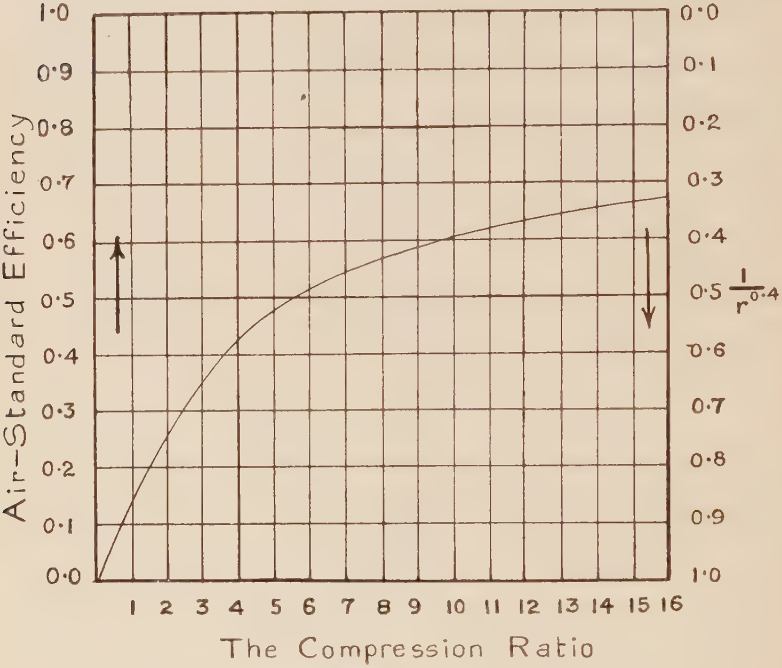

FIG. 32.

On the right-hand side are values of $\frac{1}{r^{0.4}}$, which are useful in determining the effect of "modifying factors" referred to below. This scale reads downwards.

The early engines on this basis were not nearly so efficient as the modern types of gas engines, though, when an explosive substance is compressed, the maximum efficiency which can be reached is limited by the spontaneous ignition point of the mixture. In the Otto engine of 1876, p. 16, the compression ratio was about 2·66, so that the ideal efficiency from the curve was about $32\frac{1}{2}$ per cent. In the modern gas engine shown in Fig. 10, p. 25, $r=5\frac{1}{4}$, and the ideal efficiency is therefore 48 per cent.

The Clerk Two-Stroke Engine of 1880.—The ideal card for this type of engine is shown in Fig. 33.

The working substance is pumped in during the portion of the in stroke represented by 1 to 2. The Otto cycle is then followed until the point 5, where the exhaust port is again uncovered on the outward stroke. Exhaust is assumed to take place at constant volume from 5 to 2 and then at constant pressure outside the cycle from 2 to 1. In practice it follows more nearly the dotted line 5 to 1.

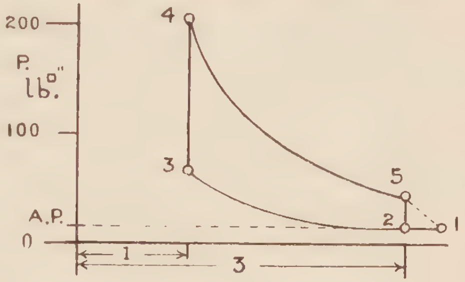

FIG. 33.

The ideal efficiency is therefore the same as the constant-volume four-stroke engine of Otto, and may be written

$$\eta = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7c)$$

The Atkinson "Cycle" Engine of 1886.—This is the first interesting attempt to improve the efficiency of the cycle by extended expansion. Suction takes place (outside the cycle) from 1 to 2. Compression is adiabatic from 2 to 3, and firing is at constant volume from 3 to 4. Expansion, which is also adiabatic, is not stopped at 5*a*, but carried on or "extended" to a point 5, which is of greater volume than the beginning of compression. Exhaust takes place from 5 to 6 at constant volume, from 6 to 2 at atmospheric pressure within the cycle and is completed from 2 to 1 outside the cycle.

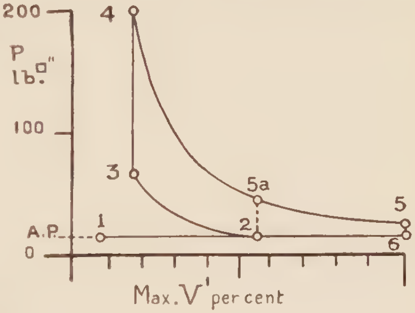

FIG. 34.

The efficiency may be computed as follows :—

Heat received at constant volume from 3 to 4 $= C_v(T_4 - T_3)$

Heat rejected at constant volume from 5 to 6 $= C_v(T_5 - T_6)$

and at constant pressure from 6 to 2 $= C_p(T_6 - T_2)$

$$\eta = 1 - \frac{C_v(T_5 - T_6) + C_p(T_6 - T_2)}{C_v(T_4 - T_3)}$$

$$= 1 - \frac{(T_5 - T_6) + \gamma(T_6 - T_2)}{(T_4 - T_3)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8a)$$

By expressing all the temperatures in terms of the suction temperature T_2, this expression may be modified as follows :—

$$T_3 = r^{\gamma-1} T_2$$

$$T_4 = aT_3 = ar^{\gamma-1} T_2$$

$$T_5=\left(\frac{V_4}{V_5}\right)^{\gamma-1}T_4=\left(\frac{V_4}{V_2}\cdot\frac{V_2}{V_5}\right)^{\gamma-1}T_4$$

$$=\left(\frac{1}{r}\cdot\frac{1}{\beta}\right)^{\gamma-1}T_4=ar^{\gamma-1}\left(\frac{1}{r\beta}\right)^{\gamma-1}T_2$$

$$=\frac{a}{\beta^{\gamma-1}}T_2$$

$$T_6=\beta T_2$$

Substituting these values in equation (8*a*) and omitting T_2, which now occurs in every term,

$$\eta=1-\frac{\left(\frac{a}{\beta^{\gamma-1}}-\beta\right)+\gamma(\beta-1)}{(ar^{\gamma-1}-r^{\gamma-1})}$$

$$=1-\frac{1}{r^{\gamma-1}}\left[\frac{\left(\frac{a}{\beta^{\gamma-1}}-\beta\right)+\gamma(\beta-1)}{a-1}\right] \quad . \quad . \quad (8b)$$

The term within the square brackets may be termed a "modifying factor," which, in this case, is less than unity, so that the ideal efficiency of the Atkinson cycle is greater than the corresponding value for the constant-volume efficiency.

Example.—The card in Fig. 9, p. 23, shows that

$$a=\frac{p_4}{p_3}=\frac{180}{50}=3{\cdot}60$$

$$\beta=\frac{V_6}{V_2}=\frac{148{\cdot}2}{100}=1{\cdot}482$$

$$r=\frac{V_2}{V_3}=\frac{100}{31{\cdot}8}=3{\cdot}143$$

The modifying factor

$$=\frac{\left(\frac{3{\cdot}6}{1{\cdot}482^{0{\cdot}4}}-1{\cdot}482\right)+1{\cdot}4(1{\cdot}482-1)}{3{\cdot}60-1}$$

$$=\frac{1{\cdot}593+0{\cdot}674}{2{\cdot}6}$$

$$=\frac{2{\cdot}267}{2{\cdot}6}=0{\cdot}873$$

If the cycle had been constant volume then

$$\eta=1-\frac{1}{3{\cdot}143^{1{\cdot}4-1}}$$

$$=1-\frac{1}{1{\cdot}581}$$

$$=1-0{\cdot}632$$

$$=36{\cdot}8 \text{ per cent.}$$

Applying the modifying factor for the Atkinson cycle

$$\begin{aligned}\eta&=1-0{\cdot}632\times0{\cdot}873\\&=1-0{\cdot}552\\&=44{\cdot}8 \text{ per cent.}\end{aligned}$$

The Humphrey Gas Pump (p. 29).—The cycle of operations of the Humphrey gas pump is an extension of the Atkinson cycle down to atmospheric pressure; that is to say, the Humphrey gas pump cycle is the constant-volume cycle with *complete* expansion.

In the ideal card the mixture is drawn in from 1 to 2 (outside the cycle of operations), adiabatically compressed from 2 to 3, exploded at constant volume between 3 and 4, and expanded adiabatically down to atmospheric pressure from 4 to 5. The exhaust therefore takes place at constant pressure from 5 to 1, of which the portion 5 to 2 is within the cycle as considered.

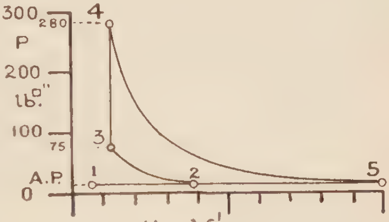

FIG. 35.

The ideal efficiency is derived as follows :—

Heat received at constant volume from 3 to 4 $=C_v(T_4-T_3)$

Heat rejected at constant pressure from 5 to 2 $=C_p(T_5-T_2)$

Hence
$$\eta=1-\frac{C_p(T_5-T_2)}{C_v(T_4-T_3)}$$
$$=1-\frac{\gamma(T_5-T_2)}{(T_4-T_3)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9a)$$

This expression may be modified, as in the case of the Atkinson cycle, by expressing all temperatures in terms of T_2.

$$T_3=r^{\gamma-1}T_2$$
$$T_4=\alpha T_3=\alpha r^{\gamma-1}T_2$$
$$T_5=\beta T_2$$
$$\therefore\ \eta=1-\frac{\gamma(\beta-1)}{(\alpha r^{\gamma-1}-r^{\gamma-1})}$$
$$=1-\frac{1}{r^{\gamma-1}}\left[\frac{\gamma(\beta-1)}{(\alpha-1)}\right] \quad . \quad . \quad . \quad . \quad (9b)$$

It should be noted that the Humphrey cycle is the special case of Atkinson cycle in which the temperatures T_5 and T_6 of the latter coincide.

Hence $T_5=T_6$ or

$$\frac{\alpha}{\beta^{\gamma-1}}=\beta \text{ (from p. 90)}$$
$$\therefore\ \alpha=\beta^{\gamma}$$

or
$$\beta=\alpha^{\frac{1}{\gamma}}$$

Hence the ideal efficiency of the Humphrey gas pump may be further simplified to

$$\eta=1-\frac{1}{r^{\gamma-1}}\left[\frac{\gamma(\beta-1)}{(\beta^{\gamma}-1)}\right]$$

or

$$1-\frac{1}{r^{\gamma-1}}\left[\frac{\gamma(\alpha^{\frac{1}{\gamma}}-1)}{(\alpha-1)}\right] \quad . \quad . \quad . \quad . \quad (9c)$$

Example.—Consider a card similar to that shown in Fig. 35, p. 91.

$$\alpha=\frac{p_4}{p_3}=\frac{280}{75}=3{\cdot}72$$

therefore $\beta=3{\cdot}72^{\frac{1}{1\cdot 4}}=2{\cdot}56$

and since $\frac{p_3}{p_2}=r^{\gamma},\quad r^{1\cdot 4}=\frac{75}{14{\cdot}7}$

whence $r=3{\cdot}2$

Therefore the modifying factor becomes by equation (9*b*)

$$\frac{1{\cdot}4(2{\cdot}56-1)}{3{\cdot}72-1}=0{\cdot}804$$

The ideal efficiency for the corresponding constant-volume cycle would be

$$\eta=1-\frac{1}{3{\cdot}2^{0\cdot 4}}=1-0{\cdot}63$$
$$=37{\cdot}0 \text{ per cent.}$$

for the Humphrey cycle

$$\eta=1-0{\cdot}63\times0{\cdot}804$$
$$=1-0{\cdot}506$$
$$=49{\cdot}4 \text{ per cent.}$$

The Diesel Oil Engine of 1897.—The well-known cycle of the four-stroke Diesel oil engine is shown in Fig. 36. Air only is

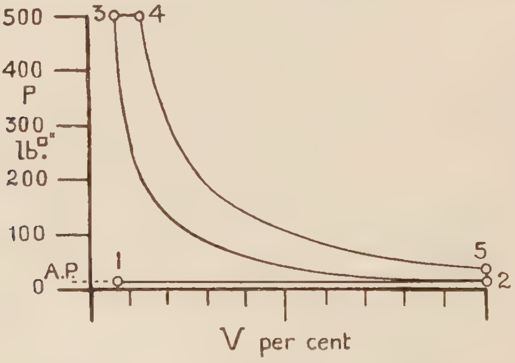

FIG. 36.

drawn in at atmospheric pressure from 1 to 2. Adiabatic compression of this air follows from 2 to 3, where $\frac{V_2}{V_3}=r$ the compression ratio.

Fuel is admitted at the end of this compression and burns (ideally) at constant pressure between 3 and 4. The remainder of the working stroke consists of adiabatic expansion from 4 to 5, whilst exhaust takes place first at constant volume from 5 to 2, and then, outside the cycle, from 2 to 1. The ideal efficiency therefore is derived as follows:

Heat received at constant pressure between 3 and $4=C_p(T_4-T_3)$
Heat rejected at constant volume between 5 and $2=C_v(T_5-T_2)$

$$\eta=1-\frac{C_v(T_5-T_2)}{C_p(T_4-T_3)}$$

$$=1-\frac{(T_5-T_2)}{\gamma(T_4-T_3)} \quad . \quad . \quad . \quad . \quad . \quad (10a)$$

But $$T_3=r^{\gamma-1}T_2$$

$$T_4=\rho T_3=\rho r^{\gamma-1}T_2$$

$$T_5=\left(\frac{V_4}{V_5}\right)^{\gamma-1}T_4=\left(\frac{V_4}{V_3}\cdot\frac{V_3}{V_5}\right)^{\gamma-1}T_4$$

$$=\rho^{\gamma-1}\times\frac{1}{r^{\gamma-1}}T_4$$

$$=\rho^{\gamma-1}\times\rho\times\frac{r^{\gamma-1}}{r^{\gamma-1}}T_2$$

$$=\rho^{\gamma}T_2$$

Hence, omitting T_2, which occurs in every term,

$$\eta=1-\frac{(\rho^{\gamma}-1)}{\gamma(\rho r^{\gamma-1}-r^{\gamma-1})}$$

$$=1-\frac{1}{r^{\gamma-1}}\left[\frac{\rho^{\gamma}-1}{\gamma(\rho-1)}\right] \quad . \quad . \quad . \quad . \quad . \quad (10b)$$

In this expression the modifying factor, within square brackets, is greater than unity, so that the attainable ideal of the Diesel cycle will be less than that deduced from a corresponding air standard. The accompanying curve (Fig. 37) gives numerical values for this modifying factor for different values of the "cut-off" ratio ρ, when the working substance is assumed to be air and $\gamma=1{\cdot}4$.

With the aid of this graph and the curve shown in Fig. 32, p. 88, the true ideal efficiency of the Diesel engine cycle is calculated as follows.

Example. $r=14$ and $\rho=2$

Then, from curve in Fig. 32,

$$\frac{1}{r^{\gamma-1}}=0{\cdot}34$$

and from curve in Fig. 37,

$$\frac{\rho^{\gamma}-1}{\gamma(\rho-1)}=1\cdot17$$

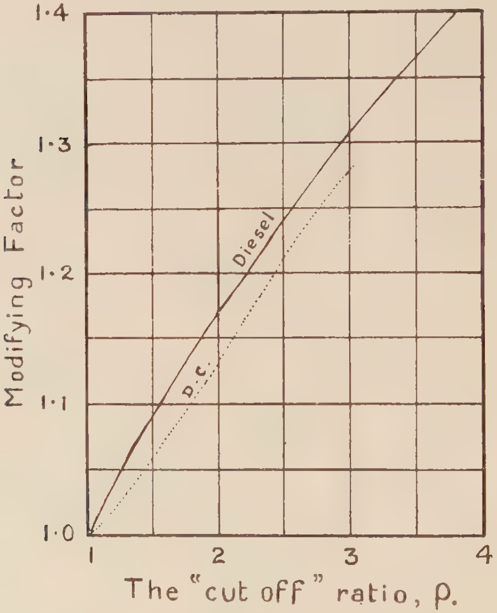

FIG. 37.

Hence the ideal efficiency

$$\begin{aligned}\eta&=1-0\cdot34\times1\cdot17\\&=1-0\cdot3978\end{aligned}$$

or 60 per cent.

This is a better standard with which to compare the performance of an actual engine than the unattainable air-standard efficiency.

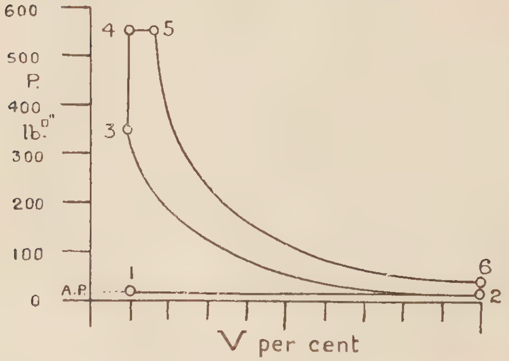

FIG. 38.

The Dual-Combustion Cycle.—The ideal efficiency of this cycle, with expansion down to the suction volume, was first derived by W. J. Walker.[1] As applied to a four-stroke engine, suction of air takes place outside the working cycle from

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[1] *Phil. Mag.*, vol. 34, 1917, p. 168.

1 to 2. The air is then compressed adiabatically from 2 to 3 and fuel is injected in such a manner as to cause it to burn partly at constant volume, from 3 to 4, and partly at constant pressure, between 4 and 5. The cycle is therefore known as the "Dual combustion" cycle, or sometimes as the "mixed" cycle. Adiabatic expansion follows from 5 to 6, and exhaust, within the cycle, at constant volume from 6 to 2. The exhaust is completed outside the diagram between 2 and 1.

To derive the efficiency for unit weight of gas :

Heat received at constant volume from 3 to $4 = C_v(T_4 - T_3)$
and at constant pressure from 4 to $5 = C_p(T_5 - T_4)$
Heat rejected at constant volume from 6 to $2 = C_v(T_6 - T_2)$

Hence $$\eta = 1 - \frac{C_v(T_6 - T_2)}{C_v(T_4 - T_3) + C_p(T_5 - T_4)} \quad . \quad . \quad . \quad (11a)$$

To reduce this expression to terms containing only the atmospheric temperature T_2 :

$$T_3 = r^{\gamma-1}T_2$$
$$T_4 = aT_3 = ar^{\gamma-1}T_2$$
$$T_5 = \rho T_4 = \rho a r^{\gamma-1}T_2$$
$$T_6 = \left(\frac{V_5}{V_6}\right)^{\gamma-1} T_5 = \left(\frac{V_5}{V_4} \cdot \frac{V_4}{V_6}\right)^{\gamma-1} T_5$$
$$= \left(\frac{V_5}{V_4} \cdot \frac{V_3}{V_2}\right)^{\gamma-1} T_5 = \rho^{\gamma-1} \cdot \frac{1}{r^{\gamma-1}} \cdot T_5$$
$$= \rho^{\gamma-1} \cdot \frac{1}{r^{\gamma-1}} \cdot \rho \cdot a \cdot r^{\gamma-1}T_2$$
$$= a\rho^{\gamma}T_2$$

Hence the ideal efficiency of the dual-combustion cycle may be written :

$$\eta = 1 - \frac{(a\rho^{\gamma} - 1)}{(ar^{\gamma-1} - r^{\gamma-1}) + \gamma(a\rho r^{\gamma-1} - ar^{\gamma-1})}$$
$$= 1 - \frac{1}{r^{\gamma-1}}\left[\frac{a\rho^{\gamma} - 1}{(a-1) + a\gamma(\rho - 1)}\right] \quad . \quad . \quad . \quad . \quad . \quad (11b)$$

This is the cycle which is followed in many modern oil engines which use solid injection. As there are three variables, the compression ratio (r), the ratio of maximum to compression pressure (a), and the "cut-off" ratio (ρ), it does not lend itself to a simple graph for the general case, but the expression is easily evaluated for any special conditions.

For example, a solid injection engine of the type shown in Fig. 23 frequently has a compression ratio (r) of 10, and a maximum pressure of about 550 lb. per sq. in. abs.

Hence for the ideal case (with constant specific heat)

$$p_3 = r^{\gamma}p_2 = 10^{1 \cdot 4} \times 14 \cdot 7$$
$$= 369 \text{ lb. per sq. in. abs.}$$

Therefore $\alpha=\frac{p_4}{p_3}=\frac{550}{369}=1{\cdot}49$, say $1{\cdot}5$

For these conditions the modifying factor within the square brackets has the values shown dotted on Fig. 37 for different values of the cut-off ratio (ρ). From the fact that this curve lies below the value for the Diesel cycle it follows that the dual-combustion cycle has a greater ideal thermal efficiency. Moreover, the shape of the curve also shows that for small values of ρ, that is to say, for a small amount of combustion at constant pressure, the ideal efficiency does not differ much from the air standard.

For example, if $\rho=1\frac{1}{3}$, the modifying factor, from the graph, equals $1{\cdot}036$.

The air-standard efficiency for a compression ratio of 10 from the graph on p. 88

$$\eta_A=1-0{\cdot}4=60 \text{ per cent.}$$

Therefore the dual-combustion efficiency

$$\eta_{DC}=1-0{\cdot}4\times1{\cdot}036$$

$$=1-0{\cdot}415=58\tfrac{1}{2} \text{ per cent.}$$

or only $1\frac{1}{2}$ per cent. less in the case under consideration.

The expression for the dual combustion (D.C.) efficiency has, however, one interesting feature in that it represents the general case which includes both the Diesel cycle and the Otto cycle.

A brief comparison of the cards will show this.

The Diesel cycle in Fig. 36 is the special case of Fig. 38, where $\alpha=1$. Substituting this value in the modifying factor of equation (11*b*) gives

$$\frac{\rho^{\gamma}-1}{\gamma(\rho-1)}$$

which is the modifying factor of the Diesel cycle.

Again, the Otto cycle of Fig. 31 is the special case of the D.C. cycle when $\rho=1$.

The modifying factor of expression (11*b*) then becomes

$$\frac{\alpha-1}{\alpha-1}=1$$

It is only necessary, therefore, to develop the thermodynamics of this type of dual-combustion cycle to include practically all the cycles on which modern types of internal-combustion engines work.

Other Forms of the Dual-Combustion Cycle.—W. J. Walker has drawn attention to other forms of the dual-combustion cycle,[1] which, at present, are chiefly of academic interest. He considers the case of dual-combustion with constant-pressure exhaust instead of constant-volume exhaust.

This cycle is shown in Fig. 39 by the numbers 2, 3, 4, 5, 8, 2.

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[1] Proc. Inst. Mech. Eng., Dec. 1920; *Automobile Engineer*, 1918, p. 27; *Engineering*, vol. cix. (1920), p. 467.

It forms a useful exercise to show, by the methods already adopted in this book, that the ideal thermal efficiency is

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{\gamma(\alpha^{\frac{1}{\gamma}}\rho - 1)}{(\alpha-1)+\alpha\gamma(\rho-1)}\right] \quad . \quad . \quad . \quad . \quad (12)$$

When $\rho=1$, this cycle becomes that of the Humphrey pump on p. 91, where the modifying factor is

$$\frac{\gamma(\alpha^{\frac{1}{\gamma}}-1)}{\alpha-1}$$

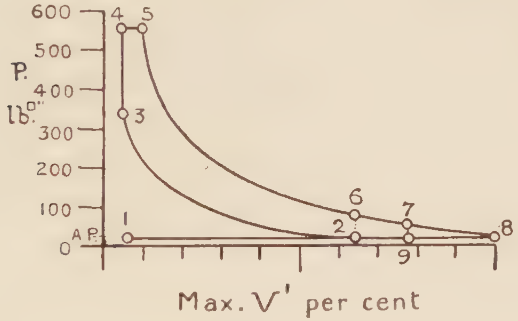

FIG. 39.

One further case may be considered here. The dual-combustion cycle with dual exhaust, as shown in Fig. 39 by the numbers

2, 3, 4, 5, 7, 9, 2.

In the *Engineering* article W. J. Walker shows that the efficiency of this cycle may be written

$$\eta = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{\left(\frac{\alpha\rho^{\gamma}}{\beta^{\gamma-1}} - \beta\right)+\gamma(\beta-1)}{(\alpha-1)+\alpha\gamma(\rho-1)}\right] \quad . \quad . \quad . \quad (13)$$

and in a further development[1] he traces a connection between this cycle and that of the Uniflow steam engine, and shows that theoretically (as well as practically) it may have a maximum efficiency when the compression ratio is about 10.

It may be noted that equation (13) becomes that of the Atkinson cycle on p. 90, when $\rho=1$.

The Dual-Combustion Cycle with Variable Specific Heats.—It will be seen in Chapter VII. that the specific heats of a gaseous mixture may be assumed to vary according to the following linear laws :

$$C_p = A + ST$$
$$C_v = B + ST$$

---

[1] Proc. Manchester Lit. and Phil. Soc., vol. 65, 1921, No. 2.

In this case, using the notation of Fig. 38, the *heat received*

$$=\int_{T_3}^{T_4} C_v dT + \int_{T_4}^{T_5} C_p dT$$

$$=(T_4-T_3)\left\{B+\frac{S}{2}(T_4+T_3)\right\}+(T_5-T_4)\left\{A+\frac{S}{2}(T_5+T_4)\right\}$$

The *heat rejected*

$$=\int_{T_2}^{T_6} C_v dT=(T_6-T_2)\left\{B+\frac{S}{2}(T_6+T_2)\right\}$$

The efficiency (η') becomes

$$\eta'_{\text{D.C.}}=1-\frac{(T_6-T_2)\left\{B+\frac{S}{2}(T_6+T_2)\right\}}{(T_4-T_3)\left\{B+\frac{S}{2}(T_4+T_3)\right\}+(T_5-T_4)\left\{A+\frac{S}{2}(T_5+T_4)\right\}}$$

To reduce this expression to terms containing only the suction temperature T_2

$$T_3=T_2 r^{\gamma-1} e^{\lambda(T_3-T_2)}$$

where $\gamma=\frac{A}{B}, \lambda=\frac{S}{B}$

and e is the base of the Napierian logs.

W. J. Walker[1] points out that λ is very small, of the order of 0·0003, so that this expression for T_3 may be written to a first approximation

$$T_3=T_2 r^{\gamma-1}\{1+\lambda T_2(1-r^{\gamma-1})\}$$

Let $T_2(1-r^{\gamma-1})=D$

then

$$T_4=\alpha r^{\gamma-1} T_2(1+\lambda D)$$

$$T_5=\rho\alpha r^{\gamma-1} T_2(1+\lambda D)$$

$$T_6=\alpha\rho\ T_2\{1+\lambda T_2(1-r^{\gamma-1}+\alpha\rho r^{\gamma-1}-\alpha\rho^{\gamma})\}$$

If these values are now inserted in the efficiency expression and higher powers of λ than the first are neglected, Walker shows that the efficiency of the D.C. cycle with variable specific heats may be written

$$\eta'_{\text{D.C.}}=\eta_{\text{D.C.}}-\frac{(\alpha\rho^{\gamma}-1)\lambda T_2}{2r^{\gamma-1}\{(\alpha-1)+\alpha\gamma(\rho-1)\}}\left[\frac{2r^{\gamma-1}\alpha\rho^{\gamma}(\alpha\rho-1)}{\alpha\rho^{\gamma}-1}-(\alpha\rho^{\gamma}-1)\right.$$
$$\left.+2(r^{\gamma-1}-1)-\frac{r^{\gamma-1}\{(\alpha^2-1)+\alpha(\rho^2-1)\}}{(\alpha-1)+\gamma\alpha(\rho-1)}\right] \quad . \quad . \quad (11c)$$

where $\eta_{\text{D.C.}}$ is the corresponding efficiency developed for constant specific heats. Equation (11*b*), p. 95.

The Constant-Volume Cycle with Variable Specific Heats.—This follows directly from the above general equation (11*c*) where $\rho=1$.

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[1] *Phil. Mag.*, vol. 34 (1917), p. 170.

Then

$$\eta'_{\text{C.V.}}=\eta_{\text{C.V.}}-\frac{\lambda T_2}{2}(\alpha+1)\left(1-\frac{1}{r^{\gamma-1}}\right)$$

$$=\eta_{\text{C.V.}}\left\{1-\frac{\lambda T_2}{2}(\alpha+1)\right\} \quad . \quad . \quad . \quad . \quad . \quad . \quad (7d)$$

where $\eta_{\text{C.V.}}$ is the corresponding constant-volume efficiency with constant specific heats derived in equation (7*c*), p. 87. As far back as 1908, H. E. Wimperis [1] had deduced the following expression for the C.V. cycle with variable specific heats:

$$\eta'_{\text{C.V.}}=\eta_{\text{C.V.}}\left\{1-(1-\eta_{\text{C.V.}})\frac{S}{B}\cdot\frac{T_4+T_3}{2}\right\} . \quad . \quad . \quad (7e)$$

where T_4 is the temperature at the end of explosion and T_3 is the temperature at the end of compression. Walker [2] shows that this equation may be reduced to equation (7*d*).

The Diesel-Engine Cycle with Variable Specific Heats.—This cycle may also be derived from the general case of equation (11*c*) by putting $\alpha=1$.

$$\eta'_{\text{D}}=\eta_{\text{D}}-\frac{(\rho^{\gamma}-1)\lambda T_2}{2\gamma r^{\gamma-1}(\rho-1)}$$

$$\left[\frac{2r^{\gamma-1}\rho^{\gamma}(\rho-1)}{\rho^{\gamma}-1}-(\rho^{\gamma}-1)+2(r^{\gamma-1}-1)-\frac{r^{\gamma-1}(\rho+1)}{\gamma}\right] \quad . \quad (10c)$$

This expression is too cumbersome for practical use, although Walker shows that, for the special case where $r=8$, it may be written very approximately as

$$\eta'_{\text{D}}=\eta_{\text{D}}\left\{1-\frac{\lambda T_2}{2}(\rho+1)\right\}$$

from which he deduces [3] that the dual-combustion cycle may take the form

$$\eta'_{\text{D.C.}}=\eta_{\text{D.C.}}\left\{1-\frac{\lambda T_2}{2}(\alpha+\rho)\right\}$$

The general case for the Diesel-engine cycle with variable specific heats was first solved by S. Lees.[4] Using the same methods which were subsequently developed by Walker to obtain his expression for the D.C. cycle, Lees gets

$$\eta'_{\text{D}}=\eta_{\text{D}}-\frac{\lambda T_2}{r^{\gamma-1}(\rho-1)\gamma}$$

$$\left[\rho^{\gamma}(1+\rho r^{\gamma-1}-r^{\gamma-1}-\rho^{\gamma})+\frac{\rho^{\gamma}-1}{2}\left\{\rho^{\gamma}-1+r^{\gamma-1}\left(2-\frac{\rho+1}{\gamma}\right)\right\}\right] . \quad (10d)$$

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[1] "The Internal Combustion Engine" (1908), p. 81.

[2] Proc. Inst. Mech. Eng., 1920, p. 1304.

[3] *Phil. Mag.*, vol. 34 (1917), p. 172.

[4] "The Effect of the Variation of Specific Heat with Temperature on the Theoretical Efficiency of the Diesel Engine," by S. Lees, *Engineering*, vol. 99 (1915), p. 1. There is a misprint in Lees' expression as published in this article. The last term is given as $\frac{\rho^{\gamma-1}}{2}\{\quad\}$ instead of $\frac{\rho^{\gamma}-1}{2}\{\quad\}$

This can be shown to agree with Walker's expression (10*c*) as follows. Multiply and divide by 2 (omitting $\eta'_D = \eta_D -$)

$$\frac{\lambda T_2}{2r^{\gamma-1}(\rho-1)\gamma}$$

$$\left[2\rho^\gamma(1+\rho r^{\gamma-1}-r^{\gamma-1}-\rho^\gamma)+(\rho^\gamma-1)\left\{\rho^\gamma-1+r^{\gamma-1}\left(2-\frac{\rho+1}{\gamma}\right)\right\}\right]$$

Multiply and divide by $\rho^\gamma - 1$

$$\frac{(\rho^\gamma-1)\lambda T_2}{2r^{\gamma-1}(\rho-1)\gamma}\left[\frac{2\rho^\gamma(1+\rho r^{\gamma-1}-r^{\gamma-1}-\rho^\gamma)}{\rho^\gamma-1}+\rho^\gamma-1+r^{\gamma-1}\left(2-\frac{\rho+1}{\gamma}\right)\right]$$

The expression within square brackets becomes

$$\frac{2r^{\gamma-1}\rho^\gamma(\rho-1)}{\rho^\gamma-1}+\frac{2\rho^\gamma(1-\rho^\gamma)}{\rho^\gamma-1}+\rho^\gamma-1+2r^{\gamma-1}-\frac{r^{\gamma-1}(\rho+1)}{\gamma}$$

omitting the first and last terms, the remainder may be written

$$-2\rho^\gamma+\rho^\gamma-1+2r^{\gamma-1}$$

which becomes $\qquad -\rho^\gamma+1+2r^{\gamma-1}-2$

or $\qquad -(\rho^\gamma-1)+2(\rho^{\gamma-1}-1)$

Collecting all the terms together gives Walker's equation (10*c*)

$$\eta'_D=\eta_D-\frac{(\rho^\gamma-1)\lambda T_2}{2r^{\gamma-1}(\rho-1)\gamma}$$

$$\left[\frac{2r^{\gamma-1}\rho^\gamma(\rho-1)}{\rho^\gamma-1}-(\rho^\gamma-1)+2(\rho^{\gamma-1}-1)-\frac{r^{\gamma-1}(\rho+1)}{\gamma}\right]$$

The Relative Efficiency.—A study of the ideal thermal efficiencies of various types of cycles is chiefly useful to the designer in that he can ascertain how near any particular engine is approaching its own ideal. The way in which actual thermal efficiencies are determined will be shown later, and the ratio between the two, namely

$$\frac{\text{actual thermal efficiency}}{\text{ideal thermal efficiency}}$$

is known as the relative efficiency.

This ratio shows how much improvement is theoretically possible, and moreover enables one type of practical design to be compared with any other.

Consideration of the relative values of the ideal thermal efficiencies, on the other hand, is of no practical value. A Diesel oil engine, for instance, is lower than the corresponding Otto cycle. But it is not practical to run to such high compressions in the latter case as are possible with Diesel engines. The cycle which is usually assumed for the Otto and Langen free-piston engine can be the most efficient of all, but the engine, as has been seen, did not even approximate to the assumed cycle, and, in addition, had a very low relative efficiency. If an efficient "condensing" internal-combustion engine or gas turbine could be devised on this cycle, there might be great possibilities in store for it.

CHAPTER VI

INDICATED MEAN EFFECTIVE PRESSURES, INDICATED HORSE-POWERS, AND CONSUMPTIONS OF IDEAL CYCLES, ASSUMING CONSTANT SPECIFIC HEATS

Mean Effective Pressure.—The mean effective pressures (p_m) of any indicator diagram may be calculated from the general expression

$$p_m = \frac{\text{work done per cycle}}{\text{working volume}}$$

$$= \frac{U}{144V_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where V_w is the working volume in cu. ft., that is, the piston displacement during the working stroke and U is the work done per cycle in ft.-lbs.

In the ideal case, the work done per cycle

$$= J(H-h) \text{ ft.-lbs. per lb. of gas}$$

Hence

$$P_m = \frac{J(H-h)}{JH} \cdot \frac{JH}{V_w}$$

$$= \eta \times \frac{JH}{V_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In order to calculate mean effective pressures, therefore, all that is necessary is to multiply the thermal efficiency by the heat received (in work units) and to divide by the working volume.

It was shown in the previous chapter that practically all cycles upon which modern types of internal-combustion engines work are included in the dual-combustion cycle with expansion down to exhaust volume. The exception is the cycle of the Humphrey gas pump which expands down to atmospheric pressure. Using the notation and figures in Chapter V., the mean effective pressure for the D.C. cycle would therefore be

$$P_m = \eta_{D.C.} \times \frac{K_v(T_4-T_3)+K_p(T_5-T_4)}{V_2-V_3} \quad . \quad . \quad (3a)$$

This expression reduces as follows:

$$V_2-V_3=\frac{RT_2}{P_2}-\frac{RT_3}{P_3}=\frac{r^\gamma RT_2}{P_3}-\frac{RT_3}{P_3}$$

$$=\frac{R}{P_3}(r^\gamma T_2-T_3)$$

$$=\frac{RT_2}{P_3}(r^\gamma-r^{\gamma-1})$$

$$=\frac{T_2}{P_3}\cdot K_v(\gamma-1)(r^\gamma-r^{\gamma-1})$$

Again, from the equation on p. 95

$$K_v(T_4-T_3)+K_p(T_5-T_4)=T_2\{K_v(\alpha r^{\gamma-1}-r^{\gamma-1})+K_p(\alpha\rho r^{\gamma-1}-\alpha r^{\gamma-1})\}$$

Therefore

$$P_m=\eta_{\text{D.C.}}\times\frac{r^{\gamma-1}P_3\{(\alpha-1)+\alpha\gamma(\rho-1)\}}{(\gamma-1)(r^\gamma-r^{\gamma-1})}$$

$$=\eta_{\text{D.C.}}P_3\frac{(\alpha-1)+\alpha\gamma(\rho-1)}{(\gamma-1)(r-1)}\quad\dots\dots\quad(3b)$$

In this expression the pressures are shown in lb. per sq. ft. (abs.); but the same equation would hold if both the pressures were taken as lb. per sq. in. (abs.).

For the Otto or constant-volume (C.V.) cycle $\rho=1$ and

$$p_m=\eta_{\text{C.V.}}p_3\frac{(\alpha-1)}{(\gamma-1)(r-1)}\quad\dots\dots\quad(4)$$

For the Diesel-engine (D) cycle $\alpha=1$ and

$$p_m=\eta_{\text{D}}p_3\frac{\gamma(\rho-1)}{(\gamma-1)(r-1)}\cdot\quad\dots\dots\quad(5)$$

In these expressions, the pressure (p_3) is that at the end of compression. Some authorities prefer to use the suction pressure (p_2). Their expressions can easily be evaluated from the above when it is remembered that, for the ideal case

$$p_3=r^\gamma p_2$$

By a similar analysis it can be shown that the mean effective pressure for the D.C. cycle with complete expansion

$$p_m=\eta p_3\frac{(\alpha-1)+\alpha\gamma(\rho-1)}{(\gamma-1)(\alpha^{\frac{1}{\gamma}}\rho r-1)}\quad\dots\dots\quad(6)$$

where η is the efficiency of this cycle (equation (12)), p. 97.

For the Humphrey gas pump $\rho=1$ in this expression. Hence

$$p_m=\eta_{\text{H}}p_3\frac{(\alpha-1)}{(\gamma-1)(\alpha^{\frac{1}{\gamma}}r-1)}\quad\dots\dots\quad(7)$$

Again, for the true constant-pressure cycle, which is not used nowadays, but which may be said to be the ideal case for the Brayton engine of 1872, p. 85, $\alpha=1$ and

$$p_m=\eta_B p_3 \frac{\gamma(\rho-1)}{(\gamma-1)(\rho r-1)} \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

The mean effective pressure for the dual-combustion cycle with dual exhaust, referred to in Chapter V. p. 97, may be derived as follows:

$$P_m=\eta_{13}\times\frac{K_v(T_4-T_3)+K_p(T_5-T_4)}{V_9-V_3}$$

(see Fig. 39), where η_{13} is the efficiency of this cycle as derived by Walker and quoted in formula (13), p. 97.

$$V_9-V_3=\frac{RT_9}{P_9}-\frac{RT_3}{P_3}=R\left(\frac{T_9}{P_2}-\frac{T_3}{P_3}\right)$$

$$=R\left(\frac{r^\gamma T_9}{P_3}-\frac{T_3}{P_3}\right)=\frac{R}{P_3}(r^\gamma T_9-T_3)$$

$$=\frac{RT_2}{P_3}(\beta r^\gamma-r^{\gamma-1})$$

$$=\frac{T_2}{P_3}\cdot K_v(\gamma-1)(\beta r^\gamma-r^{\gamma-1})$$

As before, p. 102, $K_v(T_4-T_3)+K_p(T_5-T_4)$

$$=T_2\{(K_v(\alpha r^{\gamma-1}-r^{\gamma-1})+K_p(\alpha\rho r^{\gamma-1}-\alpha r^{\gamma-1})\}$$

and
$$p_m=\eta_{13}\frac{T_2P_3}{T_2}\cdot\frac{r^{\gamma-1}\{K_v(\alpha-1)+\alpha K_p(\rho-1)\}}{K_v(\gamma-1)(\beta r^\gamma-r^{\gamma-1})}$$

$$=\eta_{13}P_3\frac{(\alpha-1)+\alpha\gamma(\rho-1)}{(\gamma-1)(\beta r-1)}$$

The Lenoir engine of 1860 and the Otto and Langen free-piston engine both have similar expressions for their mean effective pressure.

For the Lenoir engine

$$p_m=\eta_L\frac{K_v(T_3-T_2)}{V_5-V_2}$$

For the free-piston engine

$$p_m=\eta_{O.L.}\frac{K_v(T_3-T_2)}{V_4-V_2}$$

In the first case

$$V_5-V_2=V_2(\beta-1)$$

$$=\frac{RT_2}{P_2}(\beta-1)$$

$$=K_v(\gamma-1)(\beta-1)\frac{T_2}{P_2}$$

similarly
$$V_4-V_2=K_v(\gamma-1)(r-1)\frac{T_2}{P_2}$$

Hence for the Lenoir engine

$$p_m=\eta_{\text{I}}p_2\frac{(\alpha-1)}{(\gamma-1)(\beta-1)} \quad . \quad . \quad . \quad . \quad . \quad (9)$$

and for the free-piston engine

$$p_m=\eta_{\text{O.L.}}p_2\frac{(\alpha-1)}{(\gamma-1)(r-1)} \quad . \quad . \quad . \quad . \quad (10)$$

These formulæ for the mean effective pressure are derived, at present, for the ideal case when the working substance is supposed to have the properties of air, and the ratio of the specific heats at constant pressure and constant volume (γ) remains constant and is equal to 1·4. Moreover, compression and expansion are both assumed to be adiabatic. These three effects will all modify the absolute value of the mean effective pressure as compared with that obtained in practice, but a study of their relative values is none the less valuable.

Take, for example, the Otto-cycle formula (4)

$$p_m=\eta_{\text{C.V.}}p_3\frac{(\alpha-1)}{(\gamma-1)(r-1)} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

For the ideal case this may be written

$$p_m=\eta_{\text{C.V.}}14{\cdot}7\frac{r^{1\cdot4}(\alpha-1)}{(1{\cdot}4-1)(r-1)} \quad . \quad . \quad . \quad . \quad (4a)$$

which contains two variables, α and r.

Values of α may be plotted against values of p_m for different values of the compression ratio r.

The result is shown in the graph of Fig. 40, and this forms a convenient chart for the solution of the formula.

By adjusting p_2 and γ (or rather n) to conditions pertaining in any particular type of engine, similar curves may be constructed which will give p_m as it should occur in practice.

Indicated Horse-power.—The horse-power of any engine may be defined as

$$\frac{\text{work done per cycle}\times\text{cycles per minute}}{33000}$$

or

$$\text{H.P.}=\frac{\text{UM}}{33000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

where U=work done per cycle in ft.-lb.=J(H$-h$), and M=number of cycles per minute; but by equations (1) and (2)

$$\frac{\text{U}}{\text{V}_w}=\text{P}_m=\eta\frac{\text{JH}}{\text{V}_w}$$

Hence

$$\text{U}=\eta\text{JH}$$

and the equation for horse-power may be written

$$\text{H.P.}=\frac{\eta\text{JHM}}{33000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Again, by equation (1)

$$U = 144 V_w p_m$$

therefore $$\text{H.P.} = \frac{p_m 144 V_w M}{33000} \quad \ldots\ldots\ldots \quad (13)$$

but $$V_w = LA$$

$$= L \frac{a}{144}.$$

therefore $$\text{H.P.} = \frac{p_m L a M}{33000} \quad \ldots\ldots\ldots \quad (14)$$

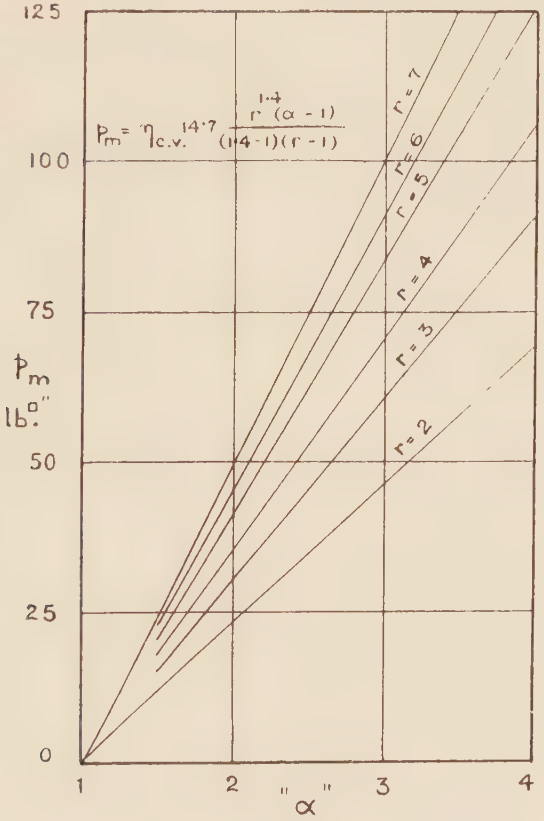

FIG. 40.

Any one of these four equations may be used to evaluate the indicated horse-power of the ideal cycle.

Consider the general case of the dual-combustion cycle with suction-volume exhaust (p. 94).

By equation (12), for instance, the heat received per lb. of working substance may be written (by analogy from equation (3*b*))

$$JH=r^{\gamma-1}K_vT_2\{(\alpha-1)+\alpha\gamma(\rho-1)\}$$

Hence $$H.P.=\eta_{D.C.}[r^{\gamma-1}K_vT_2\{(\alpha-1)+\alpha\gamma(\rho-1)\}]\frac{M}{33000} \quad . \quad (12a)$$

per lb. of working substance.

The same expression divided by v, the specific volume per lb., gives the corresponding H.P. per cubic ft. of cylinder contents.

In the ideal case

$$R \text{ for air}=96 \text{ ft.-lb. per lb. in °C.}$$

$$T_2=273^\circ \text{ C. at S.T.P.}$$

and $$K_v=\frac{R}{\gamma-1}=\frac{96}{0{\cdot}4}=240 \text{ ft.-lb. per lb.}$$

Hence $$K_v\times T_2=240\times 273=65{,}520$$

and $$H.P.=\eta_{D.C.}[r^{\gamma-1}\{(\alpha-1)+\alpha\gamma(\rho-1)\}]\frac{65520M}{33000} \quad . \quad (12b)$$

For the C.V. cycle when $\rho=1$

$$H.P.=\eta_{C.V.}[r^{\gamma-1}(\alpha-1)]\frac{65520M}{33000} \quad . \quad . \quad . \quad (12c)$$

For the Diesel cycle when $\alpha=1$

$$H.P.=\eta_{D}[r^{\gamma-1}\gamma(\rho-1)]\frac{65520M}{33000} \quad . \quad . \quad . \quad (12d)$$

In each case this value is per lb. of working substance per cycle, assumed to be air, under ideal conditioms.

For air, $v=12{\cdot}387$ cu. ft. per lb. at S.T.P.

Hence equations (12*b*), (12*c*), or (12*d*) divided by 12·387 would give the H.P. per cu. ft. of cylinder contents in each case.

Example.—An engine works on the constant-volume cycle, where $r=5$ and $\alpha=3$.

Then the ideal horse-power by formula (12*c*)

$\eta_{C.V.}=0{\cdot}47$ from chart, p. 88

$r^{\gamma-1}=5^{0{\cdot}4}=1{\cdot}904$

$\alpha-1=3-1=2$

$$H.P.=0{\cdot}47[1{\cdot}904\times 2]\frac{65520M}{33000}$$

$$=3{\cdot}545 \text{ per lb. of working substance per cycle}$$

$$=\frac{3{\cdot}545}{12{\cdot}387}=0{\cdot}287 \text{ per cu. ft. of working substance per cycle}$$

Again, equation (13) may be used. This involves a knowledge of the value of the mean effective pressure instead of the thermal efficiency.

From the evaluation of equation (3*b*) the working volume

$$V_w = \frac{T_2}{P_3} K_v(\gamma - 1)(r^\gamma - r^{\gamma-1})$$

$$= \frac{T_2}{r^\gamma 144 p_2} K_v(\gamma - 1)(r^\gamma - r^{\gamma-1})$$

$$= \frac{T_2}{144 p_2} K_v(\gamma - 1)\left(1 - \frac{1}{r}\right)$$

Therefore, from equation (13)

$$\text{H.P.} = p_m \left[\frac{T_2}{p_2} K_v(\gamma - 1)\left(1 - \frac{1}{r}\right)\right] \frac{M}{33000}$$

$$= p_m \left[\frac{RT_2}{p_2}\left(1 - \frac{1}{r}\right)\right] \frac{M}{33000} \quad . \quad . \quad . \quad . \quad . \quad (13a)$$

per lb. of working substance.

Or

$$\text{H.P.} = p_m \left[\frac{RT_2}{v p_2}\left(1 - \frac{1}{r}\right)\right] \frac{M}{33000} \quad . \quad . \quad . \quad . \quad . \quad (13b)$$

per cu. ft. of cylinder contents.

In equation (13*b*) v is the specific volume of the working substance before compression, that is to say, it represents the volume of 1 lb. of the substance under the conditions which exist at the end of the suction stroke.

The expression for the H.P. is therefore per cubic foot of the total volume of the cylinder including the clearance space. In this respect it differs from the equation (13), p. 105, from which it was derived, where the working volume was involved.

It may be noted that as equation (13*b*) contains neither α nor ρ it is common to all three cycles, dual-combustion, Diesel, or Otto, provided a suitable value of p_m is obtained.

For standard conditions (air)

$T_2 = 273°$ C. (abs.), $p_2 = 14{\cdot}7$ lb. per sq. in., $R = 96$

$\gamma = 1{\cdot}4$, and $v = 12{\cdot}387$ cu. ft.

Hence

$$\text{H.P.} = \frac{96 \times 273}{14{\cdot}7 \times 33000} p_m\left(1 - \frac{1}{r}\right)M$$

$$= 0{\cdot}05404\, p_m\left(1 - \frac{1}{r}\right) M \text{ per lb. of working substance}$$

$$= 0{\cdot}004363\, p_m\left(1 - \frac{1}{r}\right) M \text{ per cu. ft. of cylinder volume}$$

Example.—Determine the indicated horse-power per lb. (and per cu. ft.) of working substance per cycle for the constant-volume cycle when $\alpha = 3$ and $r = 5$.

From the chart, Fig. 40, on p. 105

$$p_m = 83{\cdot}5 \text{ lb. per sq. in.}$$

$$\text{H.P.} = 0{\cdot}05404 \times 83{\cdot}5(1 - \tfrac{1}{5})$$

$$= 3{\cdot}613 \text{ per lb. of working substance per cycle}$$

Alternatively

$$\text{H.P.}=0{\cdot}004363\times 83{\cdot}5\left(1-\tfrac{1}{5}\right)$$
$$=0{\cdot}2906 \text{ per cu. ft. of cylinder volume per cycle}$$

There is another important point about this equation (13*b*) which may be noted. Since it does not contain K_v but only the gas constant R, it is independent of any variations of specific heat values found in practice. It is, therefore, equally applicable to actual mixtures as it is to the ideal case of air, provided suitable adjustments are made.

For instance, it will be found on p. 124 that a suggested suitable mixture may have a gas constant of $R=99{\cdot}4$ and a specific volume (v) of 12·82 cu. ft. per lb. at S.T.P.

Then
$$\text{H.P.}=\frac{99{\cdot}4\times 273}{14{\cdot}7\times 12{\cdot}82\times 33000}\,p_m\left(1-\frac{1}{r}\right)M$$
$$=0{\cdot}004364\,p_m\left(1-\frac{1}{r}\right)M$$

per cu. ft. of cylinder volume.

The alteration in the constant is negligible and it may therefore be said that the equation derived from purely theoretical conditions represents the real state of affairs for all the three cycles usually met with in practice.

It is, however, not usual to give the total cylinder volume. A more common value is the volume swept out by the piston, or the working volume (V_w). If V_c=the clearance volume, then

$$\frac{V_w+V_c}{V_c}=r, \quad \text{whence } V_c=\frac{V_w}{r-1}$$

so that
$$V_w+V_c=V_w+\frac{V_w}{r-1}=\frac{rV_w}{r-1}$$

Hence the H.P. for any particular engine whose total cylinder volume is V_w+V_c

$$\text{H.P.}=0{\cdot}00436\,p_m\left(1-\frac{1}{r}\right)M(V_w+V_c)$$
$$=0{\cdot}00436\,p_m M\left(\frac{r-1}{r}\right)\left(\frac{r}{r-1}\right)V_w$$
$$=0{\cdot}00436\,p_m M V_w$$

Now by equation (14), p. 105

$$\text{H.P.}=\frac{p_m LaM}{33000}$$
$$=p_m\frac{L144AM}{33000}$$
$$=0{\cdot}00436\,p_m M V_w$$

This agreement in the two formulæ shows the co-relation between thermodynamical reasoning and what may be termed "mechanical" reasoning. Moreover, a close agreement between the practical and theoretical mixtures is advantageous when conceiving a hypothetical mixture which is applicable to the thermodynamical solution of practical problems.

The Consumption of Ideal Cycles.—If a fuel has a lower calorific value of Q heat units per lb. and the thermal efficiency of the internal-combustion engine is η, then the engine can only use

ηQ heat units per lb. of fuel.

Again, the number of heat units corresponding to one horse-power per hour is

$$\frac{33000}{778}\times 60 = 2546 \text{ B.Th.U.}$$

or

$$= \frac{2546}{1 \cdot 8} = 1415 \text{ C.H.U.}$$

Hence the number of pounds of fuel per H.P. per hour required by the engine is

$$W = \frac{2546}{\eta Q}$$

where Q is in B.Th.U. per lb.

Or

$$W = \frac{1415}{\eta Q}$$

where Q is in C.H.U. per lb.

The relative efficiency[1] of any engine is the ratio

$$\frac{\text{actual thermal efficiency}}{\text{ideal thermal efficiency}} = \frac{\eta_a}{\eta_i}$$

Now the actual thermal efficiency

$$\eta_a = \frac{\text{heat turned into work}}{\text{total heat available in fuel}}$$

If the actual fuel consumption of an engine is W_a lb. per H.P. per hour and the lower calorific value of the fuel is Q B.Th.U. per lb., then

$$\eta_a = \frac{2546}{W_a Q}$$

and the coefficient of performance is

$$\frac{2546}{\eta_i W_a Q}$$

But the ideal consumption of an engine

$$W_i = \frac{2546}{\eta_i Q}$$

Hence the coefficient of performance may be written as

$$\frac{W_i}{W_a} \quad \text{or} \quad \frac{\text{ideal consumption}}{\text{actual consumption}}$$

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[1] Strictly speaking the "relative efficiency" refers to the "air standard" only. "Coefficient of performance" or "Efficiency ratio" are terms which are sometimes used.

This is probably the most convenient form to use on test, as the actual consumption is always measured and the ideal consumption may be rapidly calculated for any chosen expression of the ideal efficiency. There will, of course, be two coefficients, one for indicated horse-power and one for brake horse-power, the ratio between the two being represented by the mechanical efficiency of the engine.

CHAPTER VII

VARIABLE SPECIFIC HEATS AND THE ENERGY CURVES OF GASEOUS MIXTURES

ONE of the attributes of a perfect, or ideal, gas which has been assumed up to this point is that its specific heat, either at constant pressure or at constant volume, is constant for the complete range of temperature met with in practice. At the beginning of Chapter IV. it was stated that this assumption, whilst greatly simplifying the thermodynamical relations, was not really tenable, and the way in which specific heats may vary will now be considered.

Units involved.—The specific heat of a substance represents the amount of heat required to raise unit quantity of the substance through one degree. In the case of a gas, such heating may take place under constant-volume or under constant-pressure conditions. Every gas, therefore, has two specific heats, and their symbols are distinguished by the suffixes v for constant volume and p for constant pressure. In this book a further distinction is made as to whether the amount of heat is measured in heat units (C_v and C_p) or in work units (K_v and K_p).

The relations between K_v and K_p and their connection with the energy stored in a gas have already been given in Chapter IV. p. 69 *et seq.*

It is shown there that a determination of the specific heat at constant volume is sufficient to obtain the corresponding value at constant pressure, provided the gas constant is known, and moreover that the change in internal energy is

$$E = \int C_v dT$$

The change in constant-pressure energy, or total heat as it is sometimes called, may be determined from the relation

$$I = \int C_p dT$$

From these it follows that both energy and specific heats may be expressed in the same units.

The fundamental unit which is used by the physical chemist is known as *the gramme-calorie per gramme-molecule*, and the last compound word is frequently abbreviated to *mol.*

The conception of this unit arises out of a principle first enunciated, in 1811, by one of the very earliest physical chemists and known by his name as Avogadro's hypothesis. It may be stated as follows :

"Equal volumes of all gases, under the same conditions of temperature and pressure, contain an equal number of molecules."

This hypothesis is so well established that it now amounts to a law. The weight of a molecule, or molecular weight, is therefore proportional to the weight per unit volume of a gas, and if this weight is measured in grammes, the quantity is known as the gramme-molecule or mol. The mol has a different value for each gas but, by Avogadro's law, all mols will occupy the same volume. This volume, at S.T.P., is 22·4 litres when referred to oxygen as a standard, though in practice there are slight discrepancies, due to the fact that real gases are not quite "perfect," and some authorities use the value 22·25 litres.

Now $R=\dfrac{PV}{mT}$

where m=molecular weight in grammes

Then $V=22{\cdot}4$ litres or 22,400 cu. cm.

at S.T.P., $P=1{\cdot}0333$ kg. per sq. cm.

and $T=273{\cdot}1°$ C.

so that $R=\dfrac{1{\cdot}0333\times22400}{m\times273{\cdot}1}=\dfrac{84{\cdot}8}{m}$ kg.-cm. per gramme-mol.

$=\dfrac{0{\cdot}848}{m}$ kg. metres per gramme-mol.

But $J=427$ kg.-m. per kilo-cal.

$=0{\cdot}427$ kg.-m. per gramme-cal.

Hence $mR=\dfrac{0{\cdot}848}{0{\cdot}427}=1{\cdot}986$ gramme-calories

This value, which is practically the same for all gases, is sometimes known as the universal gas constant.

But $$mR=mC_p-mC_v$$

where mC_p and mC_v are in molecular heat units

or $$mC_p-mC_v=1{\cdot}986$$

when measured in gramme-calories per gramme-molecule.

The continental engineer uses the kilogramme-molecule, and the kilo-calorie, in which case capital letters are used, *i.e.* Calorie per Mol.

Specific heats expressed in *gramme-calories per gramme* have the same numerical value as if they were expressed in *pound-calories per pound*, that is in *C.H.U. per lb.*

This value is readily derived from the fundamental unit if the molecular weight (m) of the gaseous mixture is known.

$$\text{C.H.U. per lb.}=\frac{\text{gramme-cal. per mol.}}{m}$$

If the values are required in work units, that is in *foot-pounds per pound*, then

$$\text{ft.-lb. per lb.}=1400\times\text{C.H.U. per lb.}$$

Lastly, specific heats are sometimes expressed in *foot-pounds per cubic foot*.

If the number of cu. ft. per lb. (v) of the gaseous mixture is known, then

$$\text{ft.-lb. per cu. ft.} = \frac{\text{ft.-lb. per lb.}}{v}$$

This last unit is frequently called *the volumetric heat* of a gas.

Sir Alfred Ewing shows [1] that it can be obtained direct from the molecular-heat value by the following relation:

$$1 \text{ gramme-cal. per mol} = 3{\cdot}9 \text{ ft.-lb. per cu. ft.}$$

The Gaseous Explosions Committee of the British Association used 3·96 instead of 3·9 in the above conversion, due to the fact that the standard volume of the mol was taken as 22·25 litres instead of the 22·4 litres upon which Sir Alfred Ewing's figures were based.

Methods of determining the Values of the Specific Heats of Gaseous Mixtures.—Since the energy curves of a suitable gaseous mixture can play an important part in determining the thermal efficiency of an internal-combustion engine cycle, and since such curves may be derived from suitable values of the two specific heats of the mixtures, a detailed consideration of these values is necessary.

In the first place it should be pointed out that there are two schools of thought in this connection. Some authorities seek to obtain results by synthesis, taking experimental values for each constituent gas, and building up a composite curve. This method of attack on the problem is the older of the two, and it still has many adherents at the present day. The other school says that actual properties of the constituent gases, as determined by the physical chemist in the laboratory, are not, in themselves, sufficient to predict what will happen when a mixture of the gases is functioning in an internal-combustion engine cylinder. According to this later train of thought, the only true way is to determine *apparent* specific-heat values, from data obtained from internal-combustion engines themselves, taking into account as many as possible of the other disturbing influences at work. Both schools tacitly assume that it is possible to obtain energy curves which will be suitable for any kind of internal-combustion engine, and both attempt to derive empirical formulæ for such specific heats.

There is a further diversity of opinion as to whether the formulæ should follow a linear law, such as

$$C = a + bT$$

where a and b are constants and T is the absolute temperature, or a quadratic law such as

$$C = a + bT \pm cT^2$$

where c is a further constant.

In both cases there is no theoretical justification for these types of formulæ. They merely form convenient expressions based on ascertained facts which approximate as closely as possible to the curves, and which can then be embodied in analytical reasoning.

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[1] "Thermodynamics for Engineers" (1920), p. 238.

The Synthetic Method.—This way of determining the energy curve of an explosive mixture was first put forward by the late Professor B. Hopkinson,[1] in 1908, and was elaborated by him in the first report of the Gaseous Explosions Committee of the British Association.[2] The work has formed the basis of all subsequent developments of this method, and these two papers will well repay a careful study. The results deduced will be referred to in this book as "B.A." results for short.

For the purposes of determining the values for any gaseous mixture used in internal-combustion engines, the constituent gases may be taken as those determined by accurate exhaust-gas analysis. These consist of, in the main,

Diatomic gases, such as air, nitrogen and oxygen, or carbon monoxide if combustion is incomplete.

Carbon dioxide, CO_2.

Water vapour, H_2O.

The B.A. Committee collated and reviewed the best evidence available at the time for the variation in the specific-heat values of these three groups.

An internal-energy curve was then built up with a mixture, first used by Sir Dugald Clerk, which had the following proportions :

B.A. Mixture.[3]

Diatomic gases	N_2	75	per cent.
	O_2	8	,,
	CO_2	5	,,
	H_2O	12	,,
	Total	100	,,

They comment on this curve as follows : "The committee are of the opinion that values of the energy obtained from explosion records are not subject to any very great errors on account of heat loss by conduction to the walls of the vessel, nor on account of incomplete combustion, but that they are affected by errors of quite unknown amount due, first, to heat radiated, and secondly to the want of thermal equilibrium at the time when the pressure is measured. For the purpose of testing the first of these conclusions, it is very desirable that further experiments should be made on explosions in vessels of greatly different size, but of similar form. The opinion entertained by the committee that incomplete combustion is a surface phenomenon, on which this conclusion as to the validity of the method is based, also requires further confirmation. As regards the second conclusion, further experiments on the actual amount of heat radiated by burning gas is urgently required, and also experiments to confirm or negative the effect of the nature of the wall surface upon the pressure reached in an explosion. The effect of want of thermal equilibrium can be determined up to a point by calculation; but before such calculation can be usefully made, it is desirable that further information should be obtained as to the temperature distribution after an explosion, especially in the neighbourhood of the walls."

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[1] "The Effect of Mixture Strength and Scavenging upon Thermal Efficiency," Proc. Inst. Mech. Eng., April, 1908, pp. 417 *et seq.*

[2] First Report, 1908, reproduced in Sir Dugald Clerk's book on "The Gas, Petrol and Oil Engine" (1910), vol. i. Appendix. There is a summary in Inchley's "Theory of Heat Engines" (1920), pp. 313 *et seq.*

[3] More correctly the values as given by Clerk were N_2 75·0 per cent., steam 11·9 per cent., CO_2 5·2 per cent., and O_2 7·9 per cent. See "Dictionary of Applied Physics," vol. i. p. 305.

There are various ways in which specific heats of gases may be determined in the laboratory. They may be briefly classified as:

(*a*) Determinations at constant pressure.

(*b*) Determinations at constant volume, or explosion experiments.

(*c*) Deduction from other methods such as the determination of the velocity of sound in hot gases.

(*a*) The constant-pressure experiments are usually carried out by passing the gas at atmospheric pressure through some form of heater, and then cooling it again in a calorimeter. Temperature readings are taken just before and after the gas passes through the calorimeter and the quantity of heat evolved is measured. The change in internal energy between these two temperatures can then be deduced. Experiments of this kind have been done up to a temperature of about 1400° C. The specific heat deduced from such results is the *mean* value between the two temperatures. Amongst others who used this method, Holborn and Henning,[1] and Swann [2] may be cited.

(*b*) Constant-volume or explosion experiments consist in firing a known combustible mixture of gases in a closed vessel and deducing the maximum temperature from the pressure reached after explosion. Here again *mean* values are obtained between the temperatures before and after explosion. Langen,[3] Pier, and Bjerrum,[4] amongst many others, used this method. It has been found possible to reach much higher temperatures than with the constant-pressure method, Bjerrum, for instance, carried his experiments on water-vapour to over 3000° C.; that is to say, he calculated a mean value for the specific heat of water vapour at about 1500° C.

Where such experiments give mean values, it is possible to obtain actual values at higher temperatures by the following methods:—

If the slope of the specific-heat curve is constant, that is, if it conforms to the linear law

$$C=a+bt$$

---

[1] *Annalen der Physik*, vol. 23 (1907), pp. 809 *et seq.*

[2] Proc. Royal Society, A, vol. 82 (1909).

[3] *Zeit. des Ver. deu. Ing.*, vol. 47 (1903), p. 622 *et seq.*

[4] First brought to the notice of British engineers by Tizard and Pye, *Automobile Engineer*, Feb., Mar., and April, 1921, and restated by Pye in "The Dictionary of Applied Physics," vol. i. p. 416.

then the mean specific heat C_m between two values C_2 and C_1

$$C_m = \frac{C_2 + C_1}{2}$$

or

$$C_2 = 2C_m - C_1$$

If, on the other hand, the specific heat curve is not constant, but has either a positive or a negative flexure, which may or may not follow a quadratic law, it is simpler to obtain actual values by a graphical method, assuming that a small portion of the curve is flat.

Fig. 41 shows diagrammatically the method of construction. The curve marked C_m represents the *mean* values between two temperatures recorded at the temperatures which correspond to the mean of the temperature range. For instance, Holborn and Henning are quoted in the "Dictionary of Applied Physics"[1] (called here D.A.P. for short) as obtaining a value of 9·33 for CO_2 between 0° C. and 1000° C. This value should be plotted on the curve at the mean temperature of 500° C. (point x). Through x draw a tangent to the curve and produce to the point y at 1000° C. Then y, or in this case 11·10, may be considered as the actual value of the specific heat of CO_2 at 1000° C. By repeating this construction for a number of points on the curve C_m, the actual curve marked C may be derived. This is a better method of construction than the one given by the author in an article published in 1923,[2] in which he overlooked the serious effect of curvature on the linear-law method.

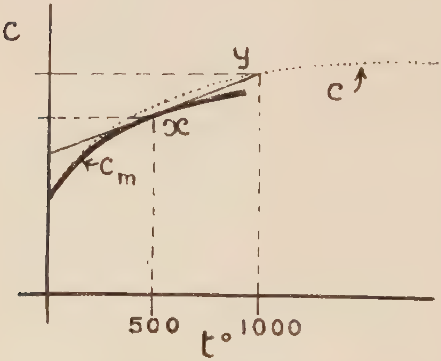

Fig. 41.

(*c*) More recently, Professor H. B. Dixon and his associates have calculated, amongst other gases, the specific heats at constant volume of N_2 and CO_2 from some very careful experiments which they carried out to determine the velocity of sound in gases at high temperatures.[3] Although the temperature range of these experiments is not nearly wide enough for the requirements of internal-combustion engineering, the authors state that their "object has been to obtain by comparative measurements the general gradients of the curves, rather than to fix the exact value at any definite point." They give as formulæ for their curves in molecular-heat units:

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[1] "Gases, Specific Heat of," by D. R. Pye. "Dictionary of Applied Physics," vol. i. (1922), p. 417.

[2] "A Note on the Effects of varying Specific Heats on the Internal-Energy Curve of a Gas-Engine Mixture," by Telford Petrie, *The Engineer*, vol. 136 (1923), p. 361.

[3] "On the Velocity of Sound in Gases at High Temperatures, and the Ratio of the Specific Heats," by H. B. Dixon, C. Campbell, and A. Parker. Proc. Roy. Soc. A., vol. 100, 1921.

Nitrogen between 0–1000° C. (ordinary)

$$4{\cdot}82+0{\cdot}00024T+0{\cdot}00000013T^2$$

Carbon dioxide between 0–700° C. (ordinary)

$$6{\cdot}30+0{\cdot}00265T+0{\cdot}0000007T^2$$

where T is in °C. absolute.

And more approximately

N_2 $4{\cdot}775+0{\cdot}00042T$
CO_2 $6{\cdot}1+0{\cdot}0028T$

Both these curves are definitely below the more recent determinations by Bjerrum and those given in the D.A.P., but their significance lies in the fact that they represent actual or instantaneous values instead of mean values as determined by previous experimenters. As will be seen later, an energy curve calculated from these values, with the addition of H_2O, gives results which agree, for all practical purposes, with a curve derived by an altogether different method by W. J. Walker (see table on p. 122).

Internal-Energy Curves by the "Synthetic Method."—The following table gives values for the actual specific heats at constant volume for every 100° C. up to 3000° C. (ordinary) in molecular heat units, that is to say, in gramme-calories per gramme-molecule. They have been drawn up by the author from a survey of all the more recent information available in May, 1924.[1] Each figure for mC_v represents the actual value between the temperature opposite which it stands and the temperature of the preceding line; *e.g.* the value of 4·92 for N_2 opposite 100° C. (ordinary) is the mean value between 0° and 100° C.

The succeeding columns represent the total increase in the internal energy from 0° C. (ordinary) in molecular heat units, and have been integrated direct, by the step-by-step method, on the assumption that the curves are flat between each successive 100° C.; *e.g.* for N_2 at 200°C.

$$\begin{aligned}\Sigma E &= 4{\cdot}92\times(100-0)+4{\cdot}98\times(200-100)\\ &= 990\end{aligned}$$

If the corresponding values for the internal energy at any temperature are required from absolute zero, the figure at the bottom of each column should be added to the value of ΣE standing opposite that temperature.

The main sources from which these figures have been drawn are the B.A. results, which have proved to be particularly valuable in determining the lower values, and the article in "The Dictionary of Applied Physics" on "Gases, Specific Heats of, at High Temperature" (vol i. pp. 415 *et seq.*), which was written by D. R. Pye. This article was published in 1922 and contains a valuable summary and survey to that date. In addition, Prof. Stodola has discussed this matter in connection with the gas turbine. His work is summarised in the fifth edition of "Dampf- und Gas-Turbinen," also published in 1922, and the pocket at the end of that book contains what, at the present time, is probably the most accurate temperature-entropy chart for the working substances of the internal-combustion engine. The greatest German authority on thermodynamics, Prof. Schüle, also gives tables for CO_2 and H_2O on pp. 62 and 63 of vol. i. of his "Technische Thermodynamik" (1921), from which it is possible to deduce directly the actual values of mC_v for every 100° C. up to 1000° C. and for every 200° C. between that value and 3000° C.

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[1] At the time this was written the author had not seen "The Specific Heat of Gases," by J. R. Partington and W. G. Shilling (1924).

TABLE I.—SPECIFIC HEATS AT CONSTANT VOLUME AND INTERNAL ENERGIES, IN MOLECULAR HEAT UNITS.

Temp.	N_2		CO_2		H_2O	
°C (ord.).	mC_v	ΣE	mC_v	ΣE	mC_v	ΣE
100	4·92	492	7·2	720	6·05	605
200	4·98	990	7·8	1,500	6·1	1,215
300	5·04	1,494	8·45	2,345	6·25	1,840
400	5·10	2,004	9·0	3,245	6·35	2,475
500	5·18	2,522	9·5	4,195	6·5	3,125
600	5·24	3,046	9·9	5,185	6·7	3,795
700	5 30	3,576	10·3	6,215	7·0	4,495
800	5·38	4,114	10·55	7,270	7·3	5,225
900	5·44	4,658	10·8	8,350	7·5	5,975
1,000	5·51	5,209	11·0	9,450	7·8	6,755
1,100	5·59	5,768	11·15	10,565	8·1	7,565
1,200	5·64	6,332	11·3	11,695	8·5	8,415
1,300	5·74	6,906	11·45	12,840	8·9	9,305
1,400	5·82	7,488	11·55	13,990	9·3	10,235
1,500	5·92	8,080	11·6	15,155	9·8	11,215
1,600	6·00	8,680	11·7	16,325	10·3	12,245
1,700	6·10	9,290	11·75	17,500	10·75	13,320
1,800	6·20	9,910	11·85	18,685	11·25	14,445
1,900	6·30	10,540	11·95	19,880	11·75	15,620
2,000	6·40	11,180	12·05	21,085	12·3	16,850
2,100	6·50	11,830	12·1	22,295	12·8	18,130
2,200	6·60	12,490	12·15	23,510	13·3	19,460
2,300	6·70	13,160	12·2	24,730	13·8	20,840
2,400	6·80	13,840	12·25	25,955	14·4	22,280
2,500	6·90	14,530	12·3	27,185	14·9	23,770
2,600	7·01	15,231	12·4	28,425	15·45	25,315
2,700	7·14	15,945	12·45	29,670	16·0	26,915
2,800	7·26	16,671	12·5	30,920	16·55	28,570
2,900	7·38	17,409	12·55	32,175	17·1	30,280
3,000	7·50	18,159	12·6	33,435	17·6	32,050
−273° to 0°	4·8	1,309	5·8	1,582	5·97	1,630

Internal-Energy Curve of Gaseous Mixture by Synthesis.—Either the column of specific heats or that of internal energy may be used, when plotted to a large scale, to determine the internal-energy curve for any particular mixture. Take, for example, the

B.A. mixture on p. 114. This mixture had 83 per cent. of diatomic gases, which may be considered as N_2, 5 per cent. of CO_2 and 12 per cent. of H_2O. Draw up a table on the following lines :

TABLE II.—SPECIFIC HEAT OF B.A. GASEOUS MIXTURE, USING THE MOST RECENT DETERMINATIONS IN MOLECULAR HEAT UNITS.

(Referred to in this book as the " Text " Mixture.)

t° C. Ordinary.	83 per cent. N_2	5 per cent. CO_2	12 per cent. H_2O	mC_v	ΣE	ΣE from $[4{\cdot}8+0{\cdot}00115T$ $5{\cdot}11+0{\cdot}00115t$
0	4·07	0·34	0·72	5·13	0	0
200	4·15	0·41	0·74	5·30	1,044	1,042
400	4·27	0·46	0·77	5·50	2,126	2,134
600	4·38	0·51	0·82	5·71	3,248	3,274
800	4·48	0·53	0·89	5·90	4,410	4,461
1000	4·66	0·55	0·92	6·13	5,612	5,693
1200	4·74	0·57	1·04	6·35	6,860	6,961
1400	4·87	0·58	1·14	6·59	8,151	8,286
1600	5·02	0·58	1·26	6·89	9,486	9,645
1800	5·19	0·59	1·38	7·16	10,875	11,057
2000	5·35	0·60	1·50	7·45	12,320	12,517
2200	5·51	0·61	1·62	7·74	13,836	14,012
2400	5·67	0·61	1·75	8·03	15,415	15,562
2600	5·86	0·62	1·88	8·36	17,057	17,159
2800	6·06	0·62	2·04	8·72	18,763	18,800
3000	6·27	0·63	2·12	9·02	20,433	20,437

The percentage columns are obtained on a slide rule, using the actual specific-heat values at the various temperatures given in the first column as read off curves plotted from Table I. The fifth column is the sum of the other three columns.

Fig. 42 shows the specific heats of Tables I. and II. plotted in molecular heat units. This should be done on a larger scale for actual use, but it will be apparent that the specific heat of a gaseous mixture as derived in the fifth column of Table II. does not increase strictly according to a linear law.

This was pointed out by Sir Alfred Ewing in his " Thermodynamics for Engineers,"[1] but the curve is sufficiently near to a straight line to enable the linear law to be used. Each student can deduce his own formula from the actual curve, but one which will be found to be sufficiently accurate for all practical purposes may be taken as follows

$$mC_v = 4{\cdot}8 + 0{\cdot}00115T$$

where T is in °C. absolute.

The sixth column of the table shows the internal energy determined from the specific-heat curve for every 100° C., as before. It could be obtained equally well by using the energy values in the first table and taking percentages direct.

The last column has been added to show the energy values as calculated from the straight line given by the above formula.

---

[1] " Thermodynamics for Engineers," by Sir Alfred Ewing (1920), p. 253.

Since T° C.$={}^\circ t+273^\circ$ C., this formula may be written

$$4{\cdot}8+0{\cdot}00115(t+273)$$
$$=5{\cdot}11+0{\cdot}00115t$$

where t is ordinary °C.

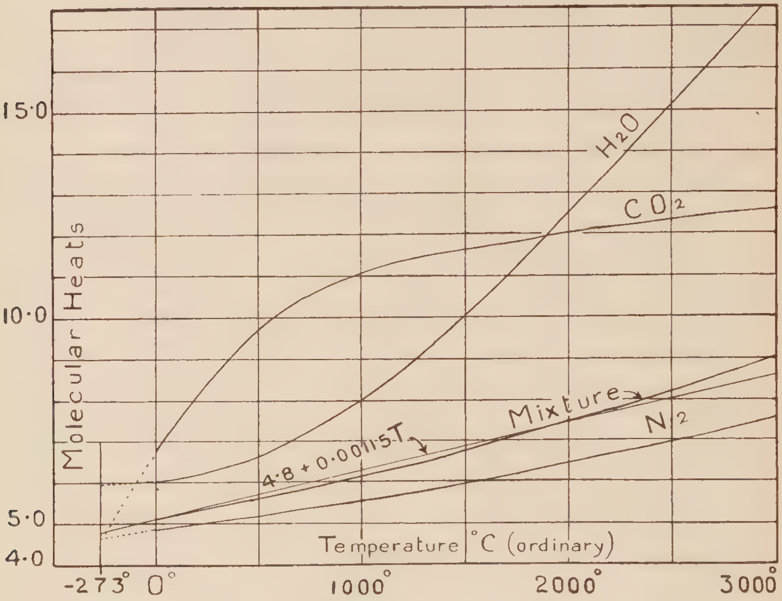

FIG. 42.—Specific Heats.

This gives an alternative method for calculating the energy increases on a straight line basis. At 1000° C., for instance,

$$E=\int_{0}^{1000}(5{\cdot}11+0{\cdot}00115t)dt$$
$$=5685 \text{ against } 5693$$

as obtained by the step-by-step process.

The "Apparent" Specific-Heat Method.—The exponents of this second method of determining the internal-energy curve of the working substance in an internal-combustion engine, hold that where such a gas is functioning in an actual cylinder there are, in addition to the variations in its specific heats, other contributory causes at work which affect the energy at any particular temperature. The result of such effects is to give an apparent value to the specific heats which may be used in theoretical calculations or comparisons. The chief difference is perhaps turbulence or the swirling movements imparted to the gases. This can be (and has been[1]) imitated to a certain extent by artificial means; but it is doubtful whether any such laboratory experiments are a true guide to the results obtained in actual engines. The most famous experiments in this latter connection are those of Sir Dugald Clerk, who carried out as far back as 1906 what are now known as his "zig-zag" experiments. These were published in the Proceedings of the Royal Society (A, vol. 77, p. 499) and elsewhere,[1] and were also reproduced in the first B.A. report. There is a very good summary of them in the D.A.P. (vol. 1, pp. 305 *et seq.*) together with a table of the internal energy calculated in ft.-lb. per cu. ft. These values may be converted to molecular heat units by dividing by 3·96 (the B.A. conversion factor), and they are included in this form in Table III.

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[1] *Cf.* Hopkinson's experiments with a fan in an enclosed explosion vessel; fifth B.A. Report (1912), p. 203.

The subject has been approached from a different standpoint by W. J. Walker.[2] He developed approximate expressions for the efficiency of different thermodynamic cycles when the variation of the specific heats according to the linear law was taken into account. He then compared the thermal efficiencies for special cases, as calculated by these expressions, with efficiencies actually obtained on test, and adjusted his constants A, B, and S, to give a maximum efficiency ratio of about 94 per cent. Walker claims that, as far as his equations go, it is possible to get efficiency ratios of over 100 per cent., if these constants, and particularly the constant S which determines the slope of the specific-heat line, exceed certain values.

In this way he derived the following formulæ for specific heat variations,

$$K_v = 220 + 0{\cdot}0623T$$
$$K_p = 317 + 0{\cdot}0623T$$

where T is in °C. (abs.) and the units are ft.-lb. per lb. of working substance.

In molecular heat units these become

$$mC_v = 4{\cdot}47 + 0{\cdot}00126T$$
$$mC_p = 6{\cdot}49 + 0{\cdot}00126T$$

In this case the difference between the two molecular heats is not exactly 1·986 because Walker worked with a hypothetical mixture which occupied 12·53 cu. ft. per lb. at S.T.P. The standard value corresponding to 1·986 would be 12·82 cu. ft. per lb.

The internal energy curve derived from this value is given in Table III.

The Derivation of Internal-Energy Curves for a Gaseous Mixture.—It will be seen from the preceding discussion that there are various ways in which an internal-energy curve may be built up or derived. There is first the synthetic method which involves the accurate determination of the specific heats at different temperatures of the various constituent gases. This is the method of the old B.A. curve, and is extended here to more recent values of the specific heats (p. 119).

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[1] "On the Limits of Thermal Efficiency in Internal-Combustion Motors," by Sir Dugald Clerk, Proc. Inst. C.E., vol. 169 (1907), pp. 121 *et seq.*

[2] "Thermodynamic Cycles of Internal-Combustion Engines," by W. J. Walker, Proc. Inst. Mech. Eng., December, 1920, p. 1306. See also "Modern Practice in Heat Engines," Petrie, 1922, pp. 199 and 200.

There is, secondly, the method of Sir Dugald Clerk, who analysed the results obtained by him from experiments in the engine itself, and then from thermodynamical relations developed in the D.A.P. article and elsewhere, derived values for the internal energy direct. There is also the third method of W. J. Walker, who worked backwards from actual engine tests and derived specific-heat equations from his own study of thermodynamics. Lastly, there is the method of estimating, in various ways, what the probable value of the specific-heat variation of a gaseous mixture is likely to be, and then deducing an energy curve by integration. This is the method advocated in a discussion on "Standards of Comparison in connection with the Thermal Efficiency of Internal-Combustion Engines," which was opened by a contribution by Prof. G. J. Wells on behalf of the Joint Committee on Tabulating the Results of Heat Engines and Boiler Trials.[1] The formula used in this paper is based on one of several given by Stodola,[2] and is as follows:

$$mC_v=4{\cdot}67+0{\cdot}001736T-0{\cdot}203\times10^{-6}T^2$$

Stodola has since modified this formula to a linear law,[3] namely,

$$mC_v=4{\cdot}67+0{\cdot}00173T$$

T being in °C. (abs.).

The following Table III. shows results obtained by these various methods. They require to be plotted to a large scale for comparison.

TABLE III.—INTERNAL ENERGIES OF A GASEOUS MIXTURE, IN GRAMME CALORIES PER MOL.

°C ordinary.	Clerk (D.A.P.).	B.A. (1908).	B.A. from Table II. ("Text" Mixture).	B.A. from Dixon's values.	Walker.
0	0	0	0	0	0
200	1,170	1,110	1.040	1,100	1,000
400	2,250	2,120	2,130	2,100	1,950
600	3,480	3,250	3,250	3,140	3,000
800	4,770	4,550	4,410	4,220	4,080
1,000	6,100	5,860	5,610	5,360	5,180
1,200	7,400	7,320	6,860	6,600	6,460
1,400	8,840	8,630	8,150	7,820	7,700
1,600	10,250	10,250	9,490	9,100	8,930
1,800	11,620	12,000	10,870	10,380	10,200
2,000	13,000	13,580	12,320	11,700	11,600
2,200	14,400	—	13,840	13,000	13,000
2,400	15,820	—	15,410	14,450	14,620
2,600	17,280	—	17,060	15,920	16,200
2,800	18,610	—	18,760	17,600	17,910
3,000	19,600	—	20,430	19,240	19,600
−273 to 0	1,300	1,240	1,350	1,200	1,250

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[1] At the time of writing (May, 1924) this committee has not yet completed its deliberations. If the course here advocated (see Proc. I.C.E., 1924) is followed, it will adjust the various methods put forward in this chapter.

[2] *Zeit. des Vereines deutscher Ing.*, vol. 56, 1912, p. 1005.

[3] "Dampf- und Gas-Turbinen," 5th edition (1922), p. 925.

The reference B.A. refers to the B.A. mixture given on p. 114. The 1908 curve is the one published in the first B.A. report and gives values which are now recognised to be too high at the higher temperatures. The second B.A. curve is the one used in this book, and referred to as the "Text" mixture. It is repeated from Table II. A curve has been added using the B.A. mixture with Dixon's (extrapolated) values for N_2 and CO_2. The H_2O value being taken from Table I. This curve is perhaps only of academic interest as the extrapolation is so considerable, but it is given here because, as far as it goes, it represents our only knowledge of actual specific heats at high temperatures as compared with the mean values over a temperature range determined by other experimenters. It is also interesting to note that it coincides, for all practical purposes, with the curve deduced by Walker from purely theoretical reasoning with the constants checked from actual engine tests.

At 2000° C. (ordinary), the point of maximum deviation, these two curves (Walker's and Dixon's) are about 10 per cent. lower than Clerk's, and 5 per cent. lower than the "text" curve of the B.A. mixture. It is, however, doubtful whether the values at that temperature have a greater accuracy than ± 5 per cent., so that, in the light of our present knowledge, the curve deduced for the B.A. mixture in Table II., may be taken to represent the mean of the values for internal energy of a gaseous mixture. It is, of course, much better, for any special case, to build up an energy curve from a knowledge of the proportions of its constituent gases. It would seem, from the above analysis, that such a procedure gives a result which is within the limits of accuracy of the present experimental data.

A "Text" Gaseous Mixture.—It would be a great convenience to have some form of standard gaseous mixture with which to compare the actual performances of internal-combustion engines, but it should be realised that such a conception can only be an arbitrary one. There are too many variations, not only in the constituents of the gases involved, but also in the amount of air present, the method of ignition, and even in the shape and size of the combustion chamber. There is also another difficulty which is not always realised. The total amount of internal energy of any gas at any definite temperature is not known or calculable; what is possible is to estimate the amount of change in such a total between any two temperatures. It therefore follows that some datum line is required from which to start calculations and charts. For thermodynamical relations, the natural starting point is the absolute zero of temperature, and so specific-heat relations in this book are given from this zero. But for charts it is often more convenient to start at freezing point or 0° C. (ordinary), and a number of authorities give energy tables with this zero. This method has therefore been followed in the previous tables, but a correction factor has been added at the bottom so that the energy relation can be converted to the absolute scale if required.

Some charts, on the other hand, are drawn up from the suction temperature of the gas, which is variously estimated at 300° C. (abs.) or 373° C. (abs.). Now the suction temperature is one of the most difficult things to measure on test, and it is doubtful whether there will ever be a concensus of opinion as to its proper value. It is therefore better to leave the choice to the individual.

T. B. Morley has pointed out[1] that most gaseous mixtures do not differ very widely from 28 in their molecular weight; this forms a very convenient number to use for a "text" mixture. Molecular-heat units are then converted to C.H.U. per lb. when divided by 28. Again, since there are 11·19 cu. ft. of oxygen in 1 lb. of the gas at S.T.P., and the molecular weight of $O_2=32$, there will be 12·82 cu. ft. per lb. in this standard mixture at S.T.P. Moreover, its gas constant

$$R=\frac{PV}{T}=\frac{144\times14{\cdot}7\times12{\cdot}82}{273}$$
$$=99{\cdot}4 \text{ in ft.-lb. per lb. units}$$
$$=\frac{99{\cdot}4\times28}{1400}=1{\cdot}987 \text{ in molecular ht. units}$$

which checks very fairly with the value of 1·986 (or sometimes 1·985) usually assigned to the universal gas constant. As has already been pointed out, the specific heat of this mixture at constant volume may have various values assigned to it; but, taking the recent values for the B.A. mixture (which approximates to a molecular weight of 28) the *molecular heats* of this "text" mixture may be taken as

$$mC_v=4{\cdot}80+0{\cdot}00115T$$

and

$$mC_p=mC_v+1{\cdot}987$$
$$=6{\cdot}787+0{\cdot}00115T$$

If a curve is required for the constant-pressure energy or total heat, 1·987*t* should be added to each value of the internal energy given in Table II. at the corresponding temperature *t*.

The corresponding values in *C.H.U. per lb.* units may be obtained by dividing the molecular heat units by 28, the assumed molecular weight of this "text" mixture.

$$C_v=0{\cdot}1714+0{\cdot}00004107T$$
$$C_p=0{\cdot}2424+0{\cdot}00004107T$$

the difference being 0·0710, which equals 1·987 divided by 28 to four decimal places.

In *ft.-lb. per lb.* units these values are multiplied by J=1400.

$$K_v=240+0{\cdot}0575T$$
$$K_p=339{\cdot}4+0{\cdot}0575T$$

the difference being the gas constant R=99·4.

Lastly, to obtain these values in *ft.-lb. per cu. ft.* divide by the specific volume, $v=12{\cdot}82$ cu. ft. Then

$$K_v=18{\cdot}72+0{\cdot}00448T$$
$$K_p=26{\cdot}46+0{\cdot}00448T$$

As a final check these last values will be found to be 3·9 times the molecular heats.

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[1] "The Limits of Efficiency of Gas Engines," *Engineering*, vol. xcviii. (1914), p. 734.

CHAPTER VIII

ENTROPY, ENTROPY DIAGRAMS AND OTHER CHARTS

Entropy.—The conception of entropy is a good deal older than its name. In his presidential address to the Physical Society in February, 1911, Professor H. L. Callendar showed how Carnot, who died in 1832, had a clear idea of the existence of such a function of state, although the name entropy was not attributed to it until its importance was more fully recognised when Clausius, in 1865, defined "entropy" as the integral of $\frac{dQ}{T}$. Such a definition is perhaps somewhat difficult to follow at first, until it is realised that the total entropy of any gas is no more known or calculable than the total internal energy. All that can be done, and all that is needful for the construction of entropy diagrams, is to calculate the change in entropy between any two chosen temperatures. The student will remember that this is what is done when determining numerical values for internal energy, and just as change in internal energy may be written

$$\Delta E = \int C_v dT$$

so change in entropy, which is usually denoted by the symbol ϕ, can be denoted by

$$\Delta\phi = \int \frac{dH}{T}$$

or for unit mass of gas

$$\Delta\phi = \int \frac{dQ}{T}$$

It follows directly from this conception that, if the transformation from one state of a gas to another is adiabatic, then $dQ = 0$ and $\Delta\phi = 0$, and there is no change of entropy. Such lines, when drawn on a temperature-entropy diagram, often called a T-ϕ diagram, are known as *isentropic* lines.

A line of this nature is shown in Fig. 43 on the ordinary indicator, or P-V, diagram marked "adiabatic." It represents the path along which a state point of a gas, whose initial state is represented on this diagram at point 1, would move when expanding or contracting according to some adiabatic law. By definition this same path would be a vertical straight line when transferred to a T-ϕ diagram with the entropy scale plotted as abscissæ. It is then called "isentropic." By a similar process of reasoning an isothermal line on the P-V diagram becomes a horizontal straight line on the T-ϕ chart.

On the other hand, lines of constant pressures or "isobars" are straight on the P-V diagram but become curved on the T-ϕ diagram; so do the corresponding lines of constant volume or isometric lines.

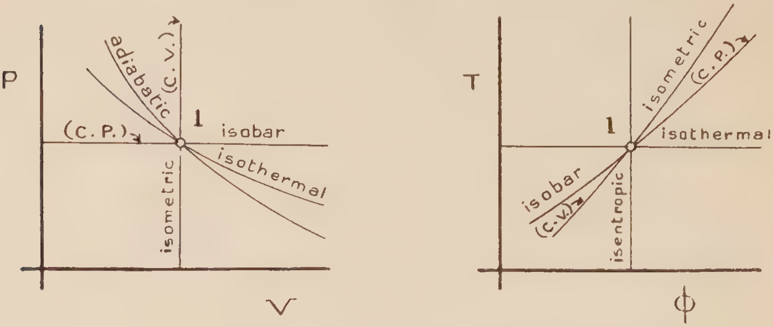

FIG. 43.

The way in which such lines may be calculated for the T-ϕ diagram will now be shown.

Joule's thermodynamic law shows that Heat taken in=Increase in Internal energy plus external work done; or, per unit quantity of gas

$$dQ=C_v dT+\frac{PdV}{J} \text{ (in heat units)}$$

dividing throughout by T

$$\frac{dQ}{T}=d\phi=C_v\frac{dT}{T}+\frac{PdV}{JT}$$

$$=C_v\frac{dT}{T}+\frac{R}{J}\cdot\frac{dV}{V} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Integrating between T_2 and T_1 and between the corresponding volumes V_2 and V_1

$$\phi_2-\phi_1=C_v \log_e\frac{T_2}{T_1}+\frac{R}{J}\log_e\frac{V_2}{V_1} \quad . \quad . \quad . \quad . \quad (2)$$

in heat units;

or $$=K_v \log_e\frac{T_2}{T_1}+R\log_e\frac{V_2}{V_1}$$

in work units.

To calculate the points of any constant-volume curve, put $V_2=V_1$ in equation (2).

Then the difference between the entropy values at any two temperatures T_2 and T_1 is

$$(\phi_2-\phi_1)_V=C_v \log_e\frac{T_2}{T_1} \quad . \quad . \quad . \quad . \quad . \quad (2a)$$

This enables a "reference" volume curve to be laid on the T-ϕ chart. In order to obtain a number of such lines, or what is usually more convenient, to construct a function volume scale showing the displacement of such a reference curve for any particular temperature, put $T_2=T_1$ in equation (2)

then $$(\phi_2-\phi_1)_T=\frac{R}{J}\log_e\frac{V_2}{V_1} \quad . \quad . \quad . \quad . \quad . \quad (2b)$$

By choosing some arbitrary value for V_1, such as 1 cu. ft., a portable scale may be constructed to show the displacement of the volume curve. To determine pressure lines on the T-ϕ diagram a similar procedure is following by putting

$$C_p-\frac{R}{J} \text{ for } C_v \text{ in equation (2).}$$

Then $$\phi_2-\phi_1=C_p\log_e\frac{T_2}{T_1}-\frac{R}{J}\left(\log_e\frac{T_2}{T_1}-\log_e\frac{V_2}{V_1}\right)$$
$$=C_p\log_e\frac{T_2}{T_1}-\frac{R}{J}\log_e\left(\frac{T_2}{T_1}\times\frac{V_1}{V_2}\right)$$
$$=C_p\log_e\frac{T_2}{T_1}-\frac{R}{J}\log_e\frac{P_2}{P_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

A reference pressure curve is then calculated from the relation

$$(\phi_2-\phi_1)_P=C_p\log_e\frac{T_2}{T_1} \quad . \quad . \quad . \quad . \quad . \quad (3a)$$

and a function scale for pressures from

$$(\phi_2-\phi_1)_T=-\frac{R}{J}\log_e\frac{P_2}{P_1} \quad . \quad . \quad . \quad . \quad . \quad (3b)$$

Constant-Volume and Constant-Pressure Curves on the Entropy Chart with Varying Specific Heats.—Equations (2) and (3) may be adapted to allow for variations in the values of specific heats which were discussed in the preceding chapter.

If the variation is assumed to follow a linear law so that

$$C_v=B+ST$$
$$C_p=A+ST$$

then equation (2) becomes

$$\phi_2-\phi_1=B\log_e\frac{T_2}{T_1}+S(T_2-T_1)+\frac{R}{J}\log_e\frac{V_2}{V_1} \quad . \quad . \quad (2c)$$

and equation (3)

$$\phi_2-\phi_1=A\log_e\frac{T_2}{T_1}+S(T_2-T_1)-\frac{R}{J}\log_e\frac{P_2}{P_1} \quad . \quad . \quad (3c)$$

If, on the other hand, a quadratic law is followed when

$$C_v=B+ST\pm S'T^2$$
$$C_p=A+ST\pm S'T^2$$

then equation (2) becomes

$$\phi_2-\phi_1=\text{B}\log_e\frac{\text{T}_2}{\text{T}_1}+\text{S}(\text{T}_2-\text{T}_1)\pm\frac{\text{S}'}{2}(\text{T}_2{}^2-\text{T}_1{}^2)+\frac{\text{R}}{\text{J}}\log_e\frac{\text{V}_2}{\text{V}_1} \qquad (2d)$$

and equation (3)

$$\phi_2-\phi_1=\text{A}\log_e\frac{\text{T}_2}{\text{T}_1}+\text{S}(\text{T}_2-\text{T}_1)\pm\frac{\text{S}'}{2}(\text{T}_2{}^2-\text{T}_1{}^2)-\frac{\text{R}}{\text{J}}\log_e\frac{\text{P}_2}{\text{P}_1} \qquad (3d)$$

Examples :—(*a*) Determine the change in entropy for the "text" mixture of Chapter VII. in C.H.U. per lb. units between 300° and 1000° C. abs. for constant volume.

The specific heat variation of this mixture in molecular heat units is assumed to be

$$m\text{C}_v=4{\cdot}8+0{\cdot}00115\text{T}$$

the molecular weight $m=28$

Hence the specific heat values in C.H.U. per lb.

$$=\text{C}_v=\frac{4{\cdot}8+0{\cdot}00115\text{T}}{28}$$

$$=0{\cdot}1714+0{\cdot}00004107\text{T}$$

Using formula (2*c*) with the omission of the last term, by analogy with formula (2*a*)

$$(\phi_2-\phi_1)_\text{V}=\text{B}\log_e\frac{\text{T}_2}{\text{T}_1}+\text{S}(\text{T}_2-\text{T}_1)$$

$$=0{\cdot}1714\log_e\tfrac{1000}{300}+0{\cdot}00004107(1000-300)$$

$$=0{\cdot}2064+0{\cdot}0287$$

$$=0{\cdot}2351$$

By calculating points for every 200° C. or so and working out the answers to four significant figures (as in Table IV., p. 130), a "reference" volume curve is obtained which may be placed on the T-ϕ chart. (See right-hand portion of Fig. 45.)

(*b*) Determine the change in entropy for the "text" mixture in C.H.U. per lb. units between 1 cu. ft. per lb. and 10 cu. ft. per lb., for constant temperature.

Using formula (2*b*), which is the same whatever formula is used for the specific-heat relation,

$$(\phi_2-\phi_1)_\text{T}=\frac{\text{R}}{\text{J}}\log_e\frac{\text{V}_2}{\text{V}_1}$$

$$\frac{\text{R}}{\text{J}}=\text{gas constant in heat units}$$

$$=\text{A}-\text{B}$$

$$=\frac{1{\cdot}987}{28}=0{\cdot}07098$$

so that

$$(\phi_2-\phi_1)_\text{T}=0{\cdot}07098\log_e\frac{\text{V}_2}{\text{V}_1}$$

If tables of hyperbolic logarithms are not to hand this may equally well be written

$$(\phi_2-\phi_1)_T=2{\cdot}303\times0{\cdot}07098\log\frac{V_2}{V_1}$$
$$=0{\cdot}1635\log\frac{V_2}{V_1}$$

Then if $V_2=10$ and $V_1=1$

$$(\phi_2-\phi_1)_T=0{\cdot}1635$$

This example shows how a scale of volumes can be made for use with the T-ϕ chart. The volume V_1 is arbitrarily chosen as 1 cu. ft. The logarithm of V_2 is then multiplied by the gas constant in heat-units and the results are plotted along the evenly divided entropy scale of the chart. If these divisions are placed on a separate strip then the portable scale so formed may be used to obtain the location of any desired volume from the single reference volume curve drawn on the chart.

A graphical method of doing this consists in drawing a line at about 30° downwards from the zero point of entropy on the T-ϕ chart, and sub-dividing this line into a logarithmic scale by means of the top divisions on a 10″-slide-rule. If the division 10 is joined to 0·1635 on the entropy scale, the other divisions follow by drawing a series of parallel lines to right and left.

This is more convenient and makes for greater accuracy than drawing a large number of volume curves on the chart. The same methods may be used to determine a reference pressure curve from the relations

$$(\phi_2-\phi_1)_P=A\log_e\frac{T_2}{T_1}+S(T_2-T_1). \quad . \quad . \quad . \quad (2c)$$

and an operative scale made from

$$(\phi_2-\phi_1)_T=-\frac{R}{J}\log_e\frac{P_2}{P_1}$$

The negative sign indicates that this scale is to be plotted in the opposite direction to that of the volume scale. If it is desired to co-ordinate both constant-volume and constant-pressure curves on the T-ϕ chart (for the D.C. cycle, for instance), these two curves must cross each other at a temperature which complies with the usual relation

$$T=\frac{PV}{R}$$

The temperature-entropy chart for modern gaseous mixtures has been developed on these lines by W. Alexander[1] and W. J. Goudie.[2] In both these cases the values for variable-specific heats were those derived by T. B. Morley from Stodola's quadratic equation. In C.H.U. per lb. units these are taken as

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[1] "An Energy Diagram for Gas Mixture and Some of its Uses," by William Alexander, Proc. Inst. Mech. Eng., May, 1917, pp. 333 to 356.

[2] "The Use of Nomograms in the Determination of Thermal Efficiencies of Internal-Combustion Engines, and Coefficients of Performance of Vapour Compression Refrigerators," by William J. Goudie, April, 1917, Institution of Eng. and Shipbuilders in Scotland, vol. 60, pp. 288–344. See also an earlier paper by

$$C_v = 0{\cdot}167 + 0{\cdot}000062T - 0{\cdot}725 \times 10^{-8}T^2$$
$$C_p = 0{\cdot}2377 + 0{\cdot}000062T - 0{\cdot}725 \times 10^{-8}T^2$$

The zero of entropy is also taken in both cases as 300° C. (abs.) and the maximum temperature at about 2200° C. (abs.).

The only difference between the two, as far as entropy calculations are concerned, is that Alexander works with a gaseous mixture which follows the characteristic law of $PV=99T$, whereas Goudie has chosen $PV=98T$. This variation should not affect the shape of any particular volume or pressure curve on the entropy chart, but it will alter the relative spacing between one curve and another.

The values used by Alexander and Goudie correspond to a universal gas constant of 1·978 and 1·960, respectively, for a gas with an average molecular weight of 28. This round figure of 28 is so convenient for conversion from molecular heat units to C.H.U. per lb., that it seems worth while making an effort to retain it when considering a "text" mixture. As the universal gas constant is taken by the physical chemist as 1·985 or 1·986, this gives R, in work units, the value of

$$\frac{1{\cdot}986 \times 1400}{28} = 99{\cdot}3$$

actually 99·4 is taken (see p. 124) for this "text" gas mixture.

The following table gives the values of $(\phi_2 - \phi_1)_V$ for plotting a reference volume curve which starts from 300° C. (abs.).

TABLE IV.—VALUES OF $(\phi_2 - \phi_1)_V$ FOR PLOTTING REFERENCE VOLUME LINES ON T-ϕ CHART (C.H.U. PER LB.).

T° C. (abs.).	(*a*) Goudie.	(*b*) Alexander.	(*c*) "Text" Mixture.	(*d*)	(*e*)
300	0	0	0		
500	0·0970	0·0968	0·0957		
700	0·1645	0·1645	0·1620		
800	0·1928	0·1928	0·1871		
1,000	0·2431	0·2420	0·2351		
1,200	0·2826	0·2814	0·2742		
1,400	0·3186	0·3182	0·3091		
1,600	0·3516	0·3516	0·3401		
1,800	0·3776	0·3782	0·3687		
2,000	0·4081	0·4078	0·3949		
2,200	0·4327	0·4330	0·4195		
2,500	—	—	0·4372		
2,750	—	—	0·4630		
3,000	—	—	0·4813		

The column marked (*a*) represents the values as calculated by Goudie and given in his paper. The next column (*b*) was estimated from the large chart prepared by Alexander and deposited

W. J. Goudie on "Gas Engine Temperature-Entropy Diagram," Proc. Inst. Mech. Eng., February, 1908, pp. 173 *et seq.*

in the library of the Institution of Mechanical Engineers. This chart measures 24″ × 19″. The absolute-temperature scale is 1″ = 100° C., and the entropy scale is 4″ = 0·10. Volume lines are drawn upon it which range from 0·4 to 70 cu. ft., and pressure lines between 8·0 and 3000 lb. per sq. in. A scale of total heat (from 300° C.) is added as well as a scale of internal energy which starts from 417° C. (abs.). Both these scales are in C.H.U. per lb. units. It will be noted that, within the accuracy with which it is possible to read this chart, column (*a*) and column (*b*) agree, which is as it should be.

Column (*c*) has been calculated for the suggested text mixture referred to in this book (see example on p. 128). Columns (*d*) and (*e*) are left for the insertion of any other variations which may be preferred.

The scale for $(\phi_2-\phi_1)_T$ may be calculated direct from the following relations ($V_1=1$ cu. ft.) :—

$$(\phi_2-\phi_1)_T = 0{\cdot}16118 \log V_2 \text{ (Goudie)}$$
$$= 0{\cdot}1632 \log V_2 \text{ (Alexander)}$$
$$= 0{\cdot}1635 \log V_2 \text{ (“ Text ”)}$$

The difference between Goudie and Alexander is accounted for by the different characteristic law which each has assumed.

By similar methods and using the appropriate equations a reference constant-pressure curve may be calculated in order to complete the chart. This is left as an exercise for the student, as a simpler graphical method of deriving a constant-pressure curve from a constant-volume curve is given later on p. 133.

Alignment Charts for use in Conjunction with the Entropy Chart.—Several very simple nomograms or alignment charts have been devised to determine four of the six properties of a gas when two of them are known. These six properties are Pressure, Volume, Temperature, Entropy, Internal Energy, and Total Heat. Of these Entropy and Temperature have been considered in the previous paragraph.

Pressure-volume and temperature for unit quantity are related by the characteristic gas law

$$PV = RT$$

Internal energy and total heat are related to temperature as discussed in the preceding chapter.

The simplest nomogram is shown in Fig. 44, in which an even scale of temperature is set upwards from 0° to 3000° C. (abs.) on line AB. An even scale of absolute pressures, 0 to 700 lb. per sq. in. (abs.), is set downwards on the line CD. The distance x may have any convenient value. A line is then drawn to join the zero on the temperature scale to the zero on the pressure scale. A volume scale is graduated on this by means of the characteristic law.

If $$PV = 99{\cdot}4T$$
Then $$144\,pV = 99{\cdot}4T$$
or $$T = 1{\cdot}45\,pV$$

Set out a table somewhat as follows :

P	V	T
200	2	580
600	2	1740
200	10	2900
400	10	1450, etc.

It will be found that by joining P and T according to such a table, the values for V will intersect on the diagonal joining zero pressure and temperature. After this has been checked, single lines may be used to sub-divide the diagonal CB into a suitable volume scale between, say, 1 and 25 cu. ft.

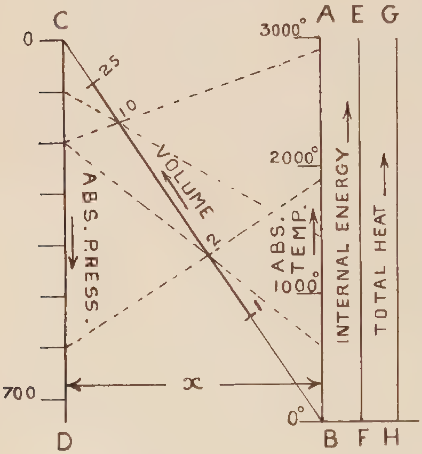

FIG. 44.—Alignment Chart Construction.

Values for internal energy and for total heat may be added direct on the vertical lines EF and GH as calculated in such tables as II. and III. in the preceding chapter. These are preferably calculated as from absolute zero and are then marked off to correspond with the temperature scale AB. If the chart is being prepared according to the specific-heat values used by Goudie and Alexander, the following table, given by Goudie in his paper, may be used for setting off values of internal energy and total heat against absolute temperature.

TABLE V.—VALUES OF TOTAL HEAT AND INTERNAL ENERGY IN C.H.U. PER LB. FROM ABSOLUTE ZERO.

Temperature °C (abs.).	Total heat.	Internal energy.	Total heat.	Internal energy.	Total heat.	Internal energy.
200	48·6	34·6	49·3	35·1		
500	126·0	91·0	126·0	91·0		
800	210·0	154·0	207·0	150·0		
1,000	266·0	196·0	262·0	192·0		
1,200	325·0	241·0	321·0	235·0		
1,500	416·0	311·0	410·0	303·0		
1,700	478·0	359·0	471·0	351·0		
2,000	578·0	438·0	567·0	425·0		
2,200	646·0	493·0	632·0	476·0		
2,400	—	—	700·0	530·0		
2,600	—	—	768·0	584·0		
2,800	—	—	839·0	641·0		
3,000	—	—	912·0	699·0		
	Goudie		“ Text ”			

The first two columns of results represent the values as calculated by Goudie from the Stodola formulæ and given in his paper.[1] The second two columns have been added for the "text" mixture suggested in this book. Two further columns are left blank for any other variations which may be preferred. If these values are plotted to a conveniently large scale, the temperatures corresponding to whole numbers in the energy curves can be read off direct and the scales added to the alignment charts shown in Fig. 44.

Combination of the Alignment Chart with the Entropy Diagram.—The temperature-entropy diagram may be conveniently combined with the alignment chart sketched in Fig. 44. It then becomes possible to determine any four of the six properties of the chosen gaseous mixture when the other two are known.

Convenient proportions for Imperial size squared paper are shown in Fig. 45. Other scales may be chosen to suit the size of the graph paper used.

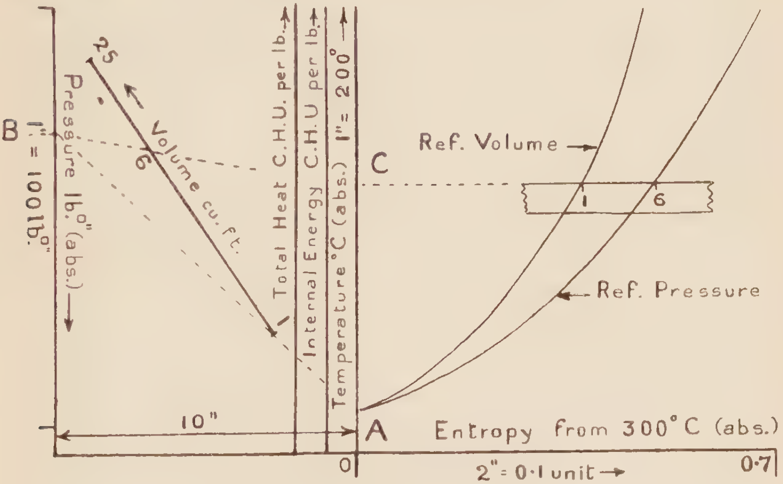
B
1" = 100 lb/□"
Pressure lb/□" (abs.)
25
6
Volume cu. ft.
Total Heat C.H.U. per lb.
Internal Energy C.H.U per lb.
Temperature °C (abs.) 1" = 200°
C
Ref. Volume
1
6
Ref. Pressure
10"
A
Entropy from 300° C (abs.)
0
2" = 0·1 unit
0·7

FIG. 45.—Entropy Chart Construction.

A reference constant-pressure curve can be added to such a chart either by calculation, as already explained, or by the following graphical method which is given by Goudie in his paper. Draw the line AB on the alignment chart, from T=300° C. (abs.) through V=1 cu. ft. This will cut the pressure scale at some value B, corresponding to the characteristic law which is adopted. Now draw a series of radial lines from this point B through various whole numbers on the volume scale. The construction is shown in the figure for $V_2=6$ cu. ft., where the line BC cuts the temperature scale at C. Through C draw a horizontal line on the T-ϕ chart across the reference volume curve. Lay the portable volume scale along this line so that 1 cu. ft. coincides with the reference volume curve and mark off a point on the chart which corresponds to the volume cut by the line BC. In the example this is 6 cu. ft. By repeating the construction, a series of points is obtained which will give a constant-pressure reference curve.

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[1] For reference, see footnote on p. 129.

It should be noted that this curve may be displaced either way at will in the T-ϕ field in a similar manner to the constant-volume curve and that a portable scale may be constructed for pressures from the relation

$$(\phi_2-\phi_1)_T=-\frac{R}{J}\log_e\frac{P_2}{P_1}$$

The negative sign means that this scale is to be constructed to read in the opposite direction to that of the portable volume scale.

Finally, to complete the usefulness of the chart, scales of internal energy and of total heat should be added alongside the scale of absolute temperature.

Examples in the use of such a chart are given in detail in both Goudie's and Alexander's papers.[1] Gas-engine indicator cards and Diesel-engine indicator cards are subdivided as accurately as is possible, and the pressure and volume points so obtained are transferred to the temperature-entropy diagram by means of the portable scales. In this way an enclosed area is obtained which represents the work done in heat units. Changes of entropy, particularly during the periods of heat reception and heat rejection, are shown up very clearly. This enables an estimate to be made of the transfer of heat between the gases and the containing walls.

Further, by surrounding this area with the ideal diagram for the limits of volume or pressure of the particular indicator card under analysis, an estimate may be made of the relative efficiency which is actually being obtained.

The difficulty of this method lies in the fact that reliable indicator cards are not easy to obtain from internal-combustion engines, chiefly on account of the much larger pressures and consequently much stronger indicator springs which are necessary as compared with steam-engine work. The pressure scale must therefore be small and difficult to estimate even when the inherent defects of the indicator itself are overcome.

For this reason many engineers do not care to use entropy charts, though it is nowadays quite possible to obtain accurate indicator cards by optical methods, which lend themselves to enlargement for analysis.[2]

Other Forms of Alignment Charts.—In addition to the simple form of alignment chart developed by Goudie and Alexander, other applications of the principles of nomography may be used. Dr. T. B. Morley has developed three charts which correlate, respectively, P, V, and T; P, ϕ, and T; and V, ϕ, and T. The methods of construction are clearly explained in his article in *Engineering*,[1] and it is therefore unnecessary to repeat them here. Logarithmic scales are used for P, V, and T, and the volume scale becomes perpendicular instead of being along an inclined axis. The actual charts given are based on the Walker values for specific heats which are

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[1] See also Chap. X. p. 175.

[2] For information regarding accurate indicating, see a group of papers published with discussions in the Proc. Inst. Mech. Eng., January, 1923, pp. 95–197.

$$C_p=0{\cdot}2265+0{\cdot}0000445T$$
$$C_v=0{\cdot}157+0{\cdot}0000445T$$

in C.H.U. per lb. units (°C. abs.); this makes $R=97$.

Scales of internal energy and of total heat are added to the temperature scales. The latter are given in both °C. (abs.) and °F. (ordinary).

A further chart is added which enables approximate values of the index n, in the relation $PV^n=\text{const.}$, to be determined graphically for any compression or expansion curve.

This chart is applicable to all gases which obey the characteristic law $PV=RT$, and the logarithmic scale enables volumes, as well as pressures, to be shown, thus rendering the chart equally suitable for steam problems.

The Walker Chart.—In a paper read before the Institution of Mechanical Engineers in 1920, W. J. Walker published a chart on a novel principle which, though avoiding entropy, is of considerable value in predicting what an actual indicator card will be like under any given conditions. The three component parts of the chart are shown in skeleton form in Fig. 46.

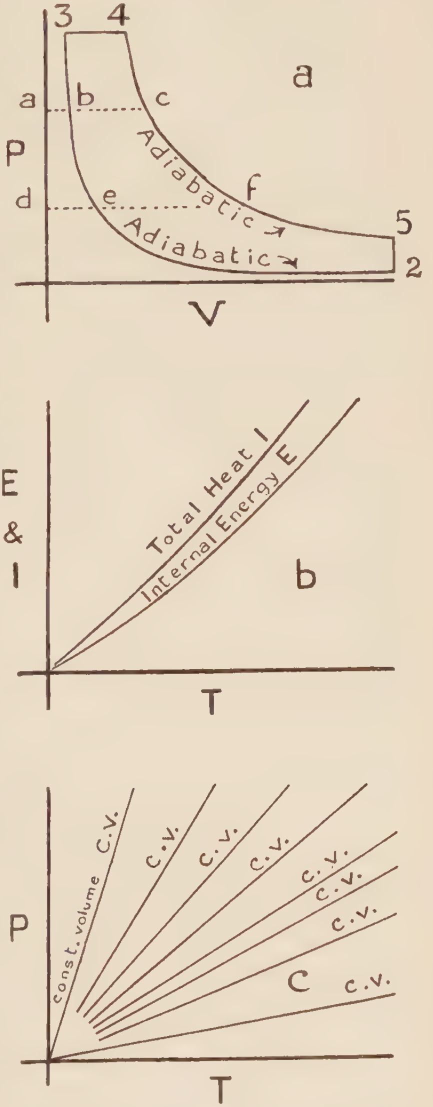

FIG. 46.—Derivation of Walker's Chart.

The groundwork consists of a series of adiabatics laid in a P-V field, these adiabatics being calculated for the specific conditions of a chosen gaseous mixture. The pressure scale may be taken in lb. per sq. in. (abs.) and the volume scale in cubic feet, or the volume scale may be given as a percentage which includes the clearance space. Compression ratios may conveniently be added to the volume scale.

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[1] "A Nomographic Chart for Gases," by T. B. Morley, *Engineering*, vol. cxii. p. 302.

The curve marked 2, 3, represents the compression of the mixture and is calculated from the point 2 for the chosen conditions at the end of the suction stroke.

The remaining curves may be added, at will, by the laws of similarity, and are used to guide the eye when drawing in the expansion curve of the particular indicator card selected. To draw them, it may be remembered that $ab : bc :: de : ef$, etc.

Superimposed on this chart are two further diagrams, one (Fig. 46*b*) connecting the absolute temperature T with the internal energy E and with the total heat I, and an intermediate diagram (Fig. 46*c*) correlating P and T for various chosen values of V. The latter are straight lines radiating from the origin in the bottom left-hand corner. The former are calculated from the two relations

$$E = \int K_v dT \quad \text{and} \quad I = \int K_p dT$$

As this chart is chiefly designed for indicator cards, it is better to express the values for internal energy and for total heat in work units, namely ft.-lb. per lb. of fuel.

The combination of these three diagrams is shown in Fig. 47, p. 140.

By this arrangement it is possible to step from an adiabatic in the pV field to a radial pT line in the pressure-temperature diagram and thence to an energy line plotted against absolute temperature.

Before going more fully into the methods by which this can be achieved it is advisable to discuss in further detail the construction of the adiabatic.

The Adiabatic Law of Gases with Variable Specific Heats.—In Chapter IV. p. 72, it is shown that when specific heats are assumed to be constant the adiabatic law may be written

$$PV^{\gamma} = \text{const.}$$

where γ is the ratio of the specific heat at constant pressure to that at constant volume. If the index has not got that value, that is to say, if a gas on compression (or expansion) follows some law such that

$$PV^{n} = \text{const.}$$

its compression is not adiabatic, since heat exchanges take place as shown on p. 75.

The compression curve 2, 3 on the Walker chart may be constructed, however, from the relation $PV^n = \text{const.}$, provided a simple value for n is considered to be accurate enough. In this case the curve will not be an adiabatic, but will represent, as near as can be estimated by this method, the actual compression curve of an indicator card taken from an engine which is using the assumed gaseous mixture. Since the value for n in expansion is often different to that assigned for compression it would make for greater accuracy to draw in the expansion curves 4, 5, etc., direct from this new value of n, rather than to develop them by similarity as in the case of real adiabatics.

It is, however, possible to derive an expression for the adiabatic law when variable specific heats are taken into account.

Assume that $C_p=A+ST$
and $C_v=B+ST$

Then for unit weight of gas

$$JdQ=JC_vdT+Pdv \quad . \quad . \quad . \quad . \quad \text{(p. 70.)}$$

and for adiabatic changes $JdQ=0$

$$\therefore J\frac{dT}{dv}=-\frac{P}{C_v} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Again $Pv=RT$

so that by differentiation

$$\begin{aligned}Pdv+vdP&=RdT\\&=J(C_p-C_v)dT\end{aligned}$$

Dividing by dv and transposing

$$\frac{P+v\dfrac{dP}{dv}}{C_p-C_v}=J\frac{dT}{dv}=-\frac{P}{C_v} \text{ from (1)}$$

$$PC_v+vC_v\frac{dP}{dv}=-PC_p+PC_v$$

$$\therefore v\frac{dP}{dv}=-P\frac{C_p}{C_v}=-P\left(\frac{A+ST}{B+ST}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$\therefore Bv\frac{dP}{dv}+STv\frac{dP}{dv}+AP+PST=0$$

Multiplying throughout by $\dfrac{dv}{Pv}$

$$B\frac{dP}{P}+ST\frac{dP}{P}+A\frac{dv}{v}+ST\frac{dv}{v}=0$$

or $$B\frac{dP}{P}+A\frac{dv}{v}+ST\left(\frac{dP}{P}+\frac{dv}{v}\right)=0$$

Let $$B\frac{dP}{P}+A\frac{dv}{v}=x$$

Then $$x+ST\left(\frac{dP}{P}+\frac{dv}{v}\right)=0$$

but $$T=\frac{Pv}{R}$$

$$\therefore\ x+\mathrm{S}\frac{\mathrm{P}v}{\mathrm{R}}\left(\frac{d\mathrm{P}}{\mathrm{P}}+\frac{dv}{v}\right)=0$$

$$x+\frac{\mathrm{S}}{\mathrm{R}}(v d\mathrm{P}+\mathrm{P}dv)=0$$

$$x+\frac{\mathrm{S}}{\mathrm{R}}d(\mathrm{P}v)=0$$

or

$$\mathrm{B}\frac{d\mathrm{P}}{\mathrm{P}}+\mathrm{A}\frac{dv}{v}+\mathrm{S}d\mathrm{T}=0$$

By integration

$$\mathrm{B}\log_e \mathrm{P}+\mathrm{A}\log_e v+\mathrm{ST}=\text{a constant}$$

or

$$\mathrm{P^B}v^{\mathrm{A}}e^{\mathrm{ST}}=\text{a constant} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

By a similar method it can be shown that, for specific heats which vary as

$$\mathrm{C}_p=\mathrm{A}+\mathrm{ST}\pm\mathrm{S'T}^2$$
$$\mathrm{C}_v=\mathrm{B}+\mathrm{ST}\pm\mathrm{S'T}^2,$$

$$\mathrm{P^B}v^{\mathrm{A}}e^{\mathrm{ST}\pm\frac{\mathrm{S'}}{2}\mathrm{T}^2}=\text{constant}$$

Equation (3) can be written

$$\mathrm{P}v^{\gamma_0}e^{\lambda\frac{\mathrm{P}v}{\mathrm{R}}}=\text{constant}$$

where $\mathrm{P}=$lb. per sq. ft. (abs.)

$v=$cu. ft. per lb.

$\gamma_0=\frac{\mathrm{A}}{\mathrm{B}}$ or ratio of specific heats when $\mathrm{T}=0$°C. (abs.)

$\lambda=\frac{\mathrm{S}}{\mathrm{B}}$

$e=$base of Napierian logs$=2{\cdot}7183$

$\mathrm{R}=$gas constant in work units

$=\mathrm{J(A-B)}$ where specific heat relations are in C.H.U. per lb.

This adiabatic equation for variable specific heats is not directly soluble, but it can be done by a process of trial and error, provided P, v, and R are known, or assumed, for one condition of the gas. The insertion of these known values, together with the constants of the chosen specific-heat relations, determines the constant in the first instance. Other values of P, and therefore of p, are then determined by trial and error for various chosen values of v.

Professor D. A. Low[1] gives another method of calculating this curve. He shows that, by writing the equation in the form

$$\mathrm{B}\log_e \mathrm{P}+\mathrm{A}\log_e v+\mathrm{ST}=\text{constant}$$

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[1] "Heat Engines" (1920), p. 528.

and then using $Pv=RT$ in the form

$$\log_e P=\log_e R+\log_e T-\log_e v$$

equations may be obtained between v and T and between P and T which enable v and P to be calculated for various assumed temperatures.

Wimperis[1] gives what is probably the neatest method of solving this equation. His solution gives the compression ratio (r) in terms of T. A tabulated example is worked out in his book for the case when

$$C_v=0{\cdot}152+0{\cdot}000075T$$
$$C_p=0{\cdot}223+0{\cdot}000075T$$

and the curve between r and t is then compared with the corresponding values for constant specific heats whose ratio, γ, is taken as 1·41.

Since $$\frac{P_1 v_1}{T_1}=\frac{P_2 v_2}{T_2}$$

or $$r=\left(\frac{v_1}{v_2}\right)=\frac{P_2}{P_1}\times\frac{T_1}{T_2}$$

$$\therefore\ P_2=P_1 r\frac{T_2}{T_1}$$

so that by assuming P_1 and T_1 for the initial condition of the gas (at the end of the suction stroke) pressures can be calculated for any compression ratio, and therefore for any volume, when T_2 is obtained by this method.

The formula derived by Wimperis is

$$\log_e r=\frac{B}{A-B}\log_e\frac{T_2}{T_1}+\frac{S}{A-B}(T_2-T_1)$$

The following table shows this adiabatic curve as calculated for a number of mixtures.

TABLE VI.—(LB. PER SQ. IN. ABS.).

Volume, per cent.	Compression ratio.	Walker, own curve.	Walker, calculated.	Wimperis, calculated.	"Text," calculated.
100·0	1·0	14	14	14	14
80·0	1·25	18	18	18	18
60·0	1·66	26	26	26	26
50·0	2·00	38	38	38	38
40·0	2·50	52	52	52	52
30·0	3·33	73	73	73	73
25·0	4·00	92	93	93	93
20·0	5·00	124	130	127	124
15·0	6·66	185	192	189	185
10·0	10·00	290	330	322	312
8·0	12·50	390	450	435	425
7·0	14·29	457	540	535	525
6·0	16·66	520	650	635	660
5·5	18·20	640	737	710	—

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[1] "The Internal-Combustion Engine," 4th edition (1922), p. 88.

The third column gives values for the Walker mixture as published by him in his 1920 paper.[1] It will be noted, by comparison with the next column, that they are not strictly adiabatic at the higher pressures. It is, however, a weakness of this method that a very slight alteration in volume here makes a considerable difference in the pressure, and the true adiabatic curve is only followed as a convenience. In the next column is given the value for the B.A. mixture as derived from Wimperis's calculations of compression ratio and temperature referred to above. The last column shows the curve which is to be expected for the " text " mixture, which is used as an example in this book. This is based on the B.A. mixture as modified by more recent values of the specific heats of nitrogen, carbon dioxide, and water vapour.

Use of the Walker Chart.—Fig. 47 shows this chart in outline as drawn by Walker. With its aid, an indicator card for any cycle in common use can be deduced, which should be found to agree very fairly with the subsequent performance of the engine under normal full load. Two examples are shown in the figure, the constant-volume cycle and the dual-combustion cycle. Other cycles, such as the variable volume cycle or the Diesel-engine cycle, are described in Walker's paper.[2]

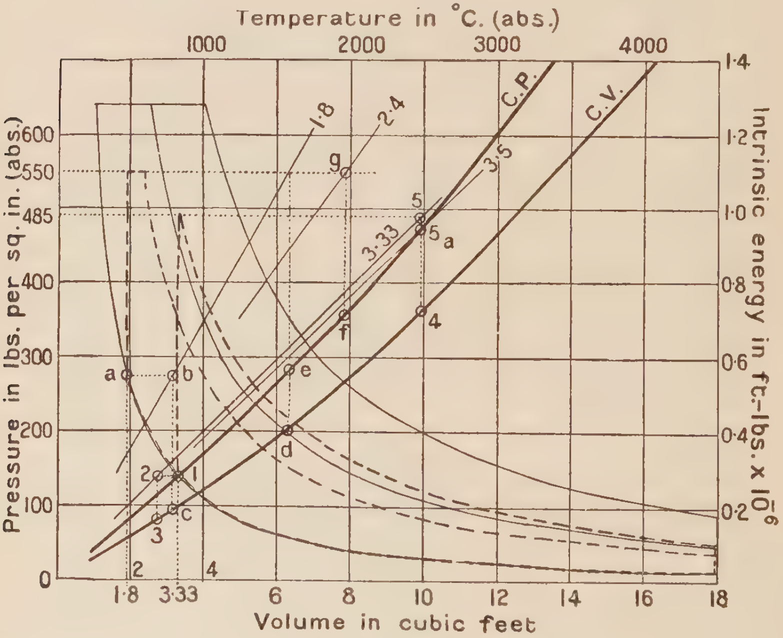

FIG. 47.—Walker's Chart.

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[1] For reference, see p. 121, footnote.

[3] Proc. Inst. Mech. Eng., December, 1920, pp. 1254 and 1305.

To draw a *constant-volume-cycle card* follow the compression adiabatic from a volume of 18 to the point marked (1). This point is determined either by the ratio of compression or by assuming the pressure at the end of compression. In the example $r=5{\cdot}4$, therefore the volume at the end of compression $=\frac{18}{5{\cdot}4}=3{\cdot}33$ cu. ft. This corresponds to a compression pressure of 135 lb. per sq. in. (abs.).

On the chart will be found a radial pT line marked 3·33 (cu. ft.). This line has been plotted to the pressure scale on the left and the temperature scale on the top. Therefore, by moving horizontally from the point (1) along the pressure line 135 lb. per sq. in. until it cuts the radial pT line of 3·33 at the point (2), the temperature can be read off on the top scale.

In the example this corresponds to 665° C. (abs.). This value can be checked from the relation $\frac{\mathrm{P}v}{\mathrm{T}}=$constant. The chart as drawn up by Walker is for 1 lb. of gas which occupies 18 cu. ft. at 373° C. (abs.) and 14·0 lb. per sq. in. Hence the required temperature

$$\mathrm{T}=\mathrm{P}v\times\frac{373}{14\times18}=1{\cdot}48\ \mathrm{P}v \text{ in °C. (abs.)}$$

$$=1{\cdot}48\times135\times3{\cdot}33=666° \text{ C. (abs.)}$$

The next step is to obtain the internal energy due to compression. The curved line marked C.V. represents this internal energy as calculated for Walker's specific-heat values. The energy scale is on the right-hand side in ft.-lb. per lb. of fuel, the figures being shown for convenience as one millionth of their actual value. As this curve is plotted against temperature, all that is necessary is to follow the temperature line of 665° C. (abs.) vertically downwards from the point (2) until it cuts this C.V. line in the point (3). The value of the internal energy is then read off on the right-hand scale. In the example it is $0{\cdot}16\times10^6$ ft.-lb. per lb. of fuel.

In order to obtain the maximum pressure, it is necessary to estimate the energy due to the heat received by the fuel during explosion. This is a comparatively simple matter if the calorific value of the gaseous mixture is known, but, as a rule, this has to be estimated from the lower calorific value of the fuel, the ratio of air to fuel, and the volumetric efficiency.

In the example, which is taken from Walker's paper, the mixture strength is given as 40·55 B.Th.U. per cu. ft. The chart is drawn up in ft.-lb. per lb. of gaseous mixture which occupies a volume of 18 cu. ft. Therefore the heat supply of the working substance equals

$$40{\cdot}55\times18\times778=0{\cdot}57\times10^6 \text{ ft.-lb. per lb.}$$

The total energy available at the end of combustion will equal

$$(0{\cdot}57+0{\cdot}16)\times10^6=0{\cdot}73\times10^6 \text{ ft.-lb. per lb.}$$

This point is shown as (4) on the C.V. line of the chart. Since the example represents a case of constant-volume explosion the maximum pressure can be found by projecting vertically upwards (or downwards) from the point (4) to the pT line 3·33 cu. ft. This gives the point (5). The pressure corresponding to this volume and temperature is then read off on the left-hand scale as 485 lb. per sq. in. (abs.). It may be here noted that if the variable-volume cycle had been worked to, instead of the constant-volume cycle and it had been assumed that the volume at the end of explosion had increased to, say, 3·5 cu. ft., then the vertical line would stop at the point (5*a*) on the pT line which represented 3·5 cu. ft. It would then be found that the maximum pressure had been reduced to 475 lb. per sq. in. (abs.).

Having determined in this way the pressures at the end of compression and at the end of explosion, the indicator card is best completed by laying over the chart a piece of tracing paper, and following the expansion curve down to exhaust pressure, as shown by chain lines in the figure. In practice the sharp peak at 485 lb. per sq. in. (abs.) would, in all probability, be rounded, and the maximum pressure would be more of the order of 420 lb. per sq. in. (abs.); but the peak value is essential in order to determine the correct expansion curve.

Two assumptions have been made in the above example with the object of simplifying the explanation in the first instance. These are the calorific value of the gaseous mixture and a volumetric efficiency of 100 per cent. This information is necessary in order to obtain a knowledge of the latent energy in the fuel. As a general rule, it has to be calculated from the lower calorific value of the fuel and the ratio of live fuel to other gases (air and exhaust-gas residue) left in the cylinder at the commencement of the compression stroke. If the volumetric efficiency is 100 per cent., which means, if all the exhaust-gas residue is swept out and only live fuel and pure air are left to fill the cylinder, then the calorific value of the mixture will be the lower calorific value (Q) of the fuel divided by $x+1$, where x is the air to fuel ratio, or

$$\text{heat supply per cylinder contents} = \frac{Q}{x+1}\ \text{C.H.U.}$$

If, on the other hand, there is a volumetric efficiency of y per cent. then the heat supply will be reduced and its value will only be

$$\frac{yQ}{x+1}\ \text{C.H.U.}$$

It so happens that Walker chose his example from a paper by W. A. Tookey,[1] where further particulars are available. The calorific value of the gas used was 500 B.Th.U. per cu. ft. and the volumetric efficiency obtained by means of a light-spring card was 87·6 per cent. Hence the air supply of this particular example can be obtained from the relation

$$40{\cdot}55 = \frac{0{\cdot}876 \times 500}{x+1}$$

whence the true air to fuel ratio is 9·8 to 1.

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[1] "Commercial Tests of Internal-Combustion Engines," W. A. Tookey, Proc. Inst. Mech. Eng., January, 1914, p. 5.

The *Diesel-engine card* may be constructed in a very similar way to that for the constant-volume cycle card. In this case the total heat or constant-pressure energy line, marked C.P. in the figure, is used for the energy relations instead of the internal-energy curve.

The procedure is otherwise the same until the value of the total energy at the end of combustion is obtained. Then, instead of moving vertically to a pT line, the point is carried to the original maximum pressure, and the corresponding volume determined from the radial pT line which is then reached.

This gives the cut-off ratio (ρ) which may then be used to complete the indicator diagram with the aid of the expansion lines.

The dual-combustion card is more elaborate. An example is worked out on Fig. 47 in letters. This card is frequently obtained from solid-injection oil engines, which have a compression ratio of about 10 and use fuel oil with a lower calorific value of about 10,000 C.H.U. per lb. The ratio of fuel to air and other gases is very high, and may be taken at about 36 to 1. This includes any allowance that need be made for volumetric efficiency.

To draw the card, follow the compression curve to a volume of 1·8 cu. ft., which corresponds to a compression ratio of 10. This is shown on Fig. 47 at the point (a). As before, move horizontally to the radial pT line for 1·8 cu. ft. (b), and then drop vertically down to the CV line (c). The internal energy due to compression is then read off the right-hand scale as $0·19 \times 10^6$ ft.-lb. The heat energy per cylinder contents with a fuel ratio of 1 to 36 and a calorific value of 10,000 C.H.U. per lb. is then

$$\frac{10,000}{36+1} \times 1400 = 0·378 \times 10^6 \text{ ft.-lb.}$$

The total energy at the end of combustion is

$$(0·19+0·378) \times 10^6 = 0·568 \times 10^6 \text{ ft.-lb.}$$

If all the fuel were burnt at constant volume, the pressure would rise to over 700 lb. per sq. in. (abs.). In practice the oil injection is controlled to give a maximum pressure of, say, 550 lb. per sq. in. (abs.).

Dropping from the pT line for 1·8 at this pressure of 550 lb. to the point d on the internal-energy line shows that the value would be $0·38 \times 10^6$ ft.-lb. per lb.[1] This would leave

$$(0·568-0·38)10^6 = 0·188 \times 10^6 \text{ ft.-lb. to be utilised at constant pressure.}$$

The energy on the constant-pressure line corresponding to $0·38 \times 10^6$ ft.-lb.[1] of internal energy is found to be $0·53 \times 10^6$ ft.-lb. (e). To this must be added the energy remaining over after the constant-volume portion of the cycle, so that the energy at the end of the combustion on the constant-pressure line is $(0·53+0·188)10^6 = 0·718 \times 10^6$ ft.-lb. This point (f) shows vertically above it at

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[1] This example, together with Fig. 47, is taken from the author's book "Modern Practice in Heat Engines" (1922), p. 252. There is, however, a mistake in the example as originally given, the figure 0·38 being shown, inadvertently, as 0·41 in the two places indicated here.

550 lb. per sq. in. (abs.) that the volume (g) on the pT line is 2·4 cu. ft. The card can then be drawn in on tracing paper as shown by the chain lines.

Indicator cards constructed, in the first instance, by means of the Walker chart may be used to determine the probable mean effective pressure which the engine under consideration should develop. Unlike the ideal steam-engine card, no diagram factor is required when using this method, but experience should be used as a guide in rounding off the sharp corners. The card will also be found of considerable use if a twisting-moment diagram of the engine is required for preliminary design. As soon as sufficient information has been obtained from the drawings to estimate the weights of the reciprocating parts, their inertia effects may be combined with the pressures shown on this card, to deduce the net resultant pressures on the piston throughout the working cycle.

CHAPTER IX

THE FORMATION OF MIXTURES REQUIRING EXTRANEOUS IGNITION

THE working substance of an internal-combustion engine is, in many ways, more complicated than that of a steam turbine or steam engine. Apart from the fact that the latter types are external-combustion engines, and, moreover, that the heat is received from one substance (the fuel) and transferred to another substance (the water) before it is introduced into the engine cylinders in the form of steam, the working substance of the internal-combustion engine is itself more complex. It is true that the bulk of it consists, in every case, of air, but the remaining portion (the fuel) may differ widely in its chemical and physical properties.

Steam is not an easy substance to handle, and its behaviour is more complicated than was at one time thought, but the fact remains that several generations of engineers and physicists have been concentrating on its problems, and the accumulated knowledge at the disposal of the steam-engine or steam-turbine designer is very considerable. Moreover, the problems of steam generation—that is, formation of heat—and its use in a working chamber are usually kept quite separate, whereas the internal-combustion engineer, as his somewhat cumbersome title implies, has to form his working substance, generate heat from it, use that heat and reject the spent products of combustion, as a rule, all in one cylinder. It does not require much imagination to realise what complications must follow from such a confined sequence of events, both in the formation of the working substance and in its behaviour in the cylinder itself.

The operations of creating, using and ejecting a suitable working medium may conveniently be divided into a number of stages.

(1) Introduction into the cylinder.
(2) Formation.
(3) Explosion and combustion.
(4) Expansion and removal from the cylinder.

These stages will have to be considered under two, or possibly under three, groups. The first group includes all types of gas engines and those oil engines, such as automobile engines, which use a controlled electric spark, or other extraneous device, to fire the mixture. In this group, a mixture of air and combustible is required which can be fired at a pre-determined moment in the cycle, but which must not pre-ignite under the heat of compression or from any other cause. The second stage mentioned above, namely, the formation of the working substance, for the most part precedes the introduction into the cylinder in this case. The distinctive feature of this first group is that the working substance is introduced at or near the beginning of compression, and that an *inflammable mixture* is therefore subject to compression.

In the second group, which will be discussed in the next chapter, air only is compressed and the fuel is introduced in various ways at or near the end of the compression stroke. This method of creating a working substance is at present confined to oil engines, though there is no inherent reason why gas under pressure might not be injected instead. The majority of these engines, such as Diesels or solid-injection engines, use no controlled extraneous firing device.

The intermediate group may be considered for the most part alongside this second group, so far as the working substance is concerned. In this group a hot-bulb, or other form of surface ignition, is used to assist in firing the mixture, but there is no control of the timing in such ignition beyond the injection of the fuel. Since air only is compressed, there is no danger of pre-ignition unless the fuel valve leaks, and thermal efficiencies may be higher than with the first group.

In addition to the above grouping there is the question of governing to be considered, in so far as it effects the formation of the working substance, and also the question of the most suitable proportions of air to fuel.

Introduction into the Cylinder.—Modern engines which employ controlled ignition all work on the constant-volume cycle, but they may be either four-stroke or two-stroke. At the present time this group includes all gas engines, whether using town gas, coke-oven gas, producer gas, or blast-furnace gas. It also includes the great majority of high-speed oil engines which work on petrol or other light spirit.

The principles underlying the introduction of the charge into the cylinder can be seen by studying the descriptions of gas engines in Chapter II. One or two obsolete types of heavy-oil engines, such as the Priestman, also come under the same category. The main point to notice is the difference between four-stroke and two-stroke operation.

Two-stroke gas engines, such as the Koerting or Oechelhauser types, require gas and air pumps to drive in the charge, whereas the four-stroke gas engine uses the working piston as a pump for three out of every four strokes. The main difficulty in connection with the introduction of the working substance into the cylinder is to form a correct estimate of its physical condition at the beginning of the compression stroke. It is very difficult to measure accurately the mean temperature of the charge at this point, and it is doubtful whether there will ever be a consensus of opinion on the matter. For the purposes of calculation such a temperature is often assumed to be 100° C. or 373° C. abs., and the pressure is taken at 14·0 lb. per sq. in. abs. This value is more easily determined than the suction temperature, as it can be read off a light-spring indicator card when due precautions are taken.

Equation (5), p. 62, Chap. IV., shows that the weight of a charge of gas

$$W=\frac{PV}{RT}$$

Consequently, for a given volume and pressure, the weight of the cylinder contents, when the inlet valves close, will be less the higher the temperature. It is a common experience in practice that the mean effective pressures are proportional to the weight of the charge after allowances are made for volumetric efficiency, so that, other things being equal, the cooler the charge at the commencement of the cycle the more power the engine will give. The mean velocity through the inlet valve may be taken as 100 ft. per second for gas engines and normal petrol engines, though for special designs of the latter for racing purposes, it may reach nearly double that amount.

Compression Ratio.—The factors governing the compression of this type of charge when once it is introduced into the working cylinder are difficult to separate out. For maximum thermal efficiency, compression should be as high as possible, but, if this compression is carried too far, pre-ignition will take place owing to the mass of the working substance reaching a temperature before the end of the compression stroke which is sufficient to self-ignite it. The engine is then said to " back-fire " or to " pink," and valuable momentum is lost. It is, however, possible to fire the charge, before this temperature is reached, by compression alone owing to the phenomenon known as detonation. This may be described as the result of some local effect, such as a hot spot, which starts an explosion wave in a mixture of air and fuel. If the expansion of gases which follows is rapid enough, it will compress remaining portions of the working substance to such a temperature that the whole is fired and an explosion occurs. The same principle is applied when firing a charge of dynamite. A small portion is compressed by percussion or otherwise, and the resulting local explosion spreads almost instantaneously to the remaining fuel.

It will be seen, therefore, that the actual compression, for any given type of fuel, is controlled by two factors: the ignition temperature of the fuel and its tendency to detonate. In both cases, the proportion of air to fuel must be such that an explosion is possible. Unfortunately, it cannot be said that the converse is true, namely, that the ignition temperature of a fuel or its liability to detonate can definitely decide the actual compression. Both factors are affected by other considerations as well, such as the maximum flame temperature and the rate of loss of heat by cooling.

The temperature of spontaneous ignition, or ignition point, is that temperature at which a fuel will ignite of its own accord when surrounded by oxygen or by air at the same temperature.[1] H. T. Tizard,[2] in a very instructive paper on " The Causes of Detonation in Internal-Combustion Engines," considers that one of the main factors governing detonation is the maximum flame temperature which any given mixture possesses. He also points out that the nature of the fuel affects its tendency to detonate. Among the liquid hydrocarbons, the aliphatics, such as the paraffin series, show the greatest tendency to detonate; whilst the aromatics, such as the benzene series, have the least tendency in that direction. Moreover, on the whole, the tendency to detonate increases with the molecular weight or with the boiling point of the fuel.[1] There are, however, exceptions to this rule, such as hydrogen, and it appears to be difficult in our present state of knowledge to predict the behaviour of a fuel in the general case.

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[1] For methods of determination of this ignition point see " Liquid Fuels," by H. Moore (1918), pp. 139 *et seq.*

[2] Trans. N. E. Coast Inst. of Engineers and Shipbuilders, vol. 37 (1921), pp. 381 *et seq.*

Current practice shows that the following compression ratios may be taken as safe average values in this connection :—

TABLE VII.—COMPRESSION RATIOS.

Type.	Fuel.	Average compression ratio.	Approximate corresponding compression press., lb. per sq. in. (abs.).
Gas Engine	Town gas . .	5½ to 1	140
,, ...	Coke oven gas .	5 to 1	125
,, ...	Producer gas . .	6½ to 1	180
,, ...	Blast furnace gas	7½ to 1	212
High-speed oil engine . .	Petrol, paraffin .	5 to 1	125
,, ,, . .	Benzol	7 to 1	195
,, ,, . .	Alcohol	9 to 1	260

For attempts to raise these compression ratios artificially by means of water injection, see Chap. XI. p. 193.

In the last section the factors which limit the amount of compression of an explosive mixture were discussed. The proportion of air to fuel in the charge is another factor which is of importance, since the calorific value of the working substance, and hence the internal energy of combustion is directly concerned. As far back as the year 1909 W. A. Tookey pointed out[2] that the influence of mixture strength upon the thermal efficiency of internal-combustion engines was of at least equal importance to that of the compression ratio. In a subsequent paper[3] he developed this point, among others, and introduced what is now known as the "Tookey factor."

This factor is defined as the ratio of the mean pressure, as shown by the positive loop of an indicator card, to the mixture strength of the charge in terms of the total cylinder contents, namely, gas, air, and residuals. The total cylinder contents are taken for this purpose as the volume of the charge at atmospheric pressure; that is to say, they represent the piston displacement multiplied by the volumetric efficiency, plus the clearance volume.

[1] See also "Oil Engines," by A. L. Bird (1923), chap. iii. p. 46.

[2] *Engineering*, vol. 88, 1909, p. 522.

[3] "Commercial Tests of Internal-Combustion Engines," by W. A. Tookey, Proc. Inst. Mech. Eng. 1914, pp. 11 *et seq.*

Tookey shows in the above paper how this factor can be calculated as follows:

If Q = Average calorific value of fuel in B.Th.U. per lb. or per cu. ft.
F = lb. or cubic feet of fuel consumed per minute.
M = Impulses per minute.
V = Effective volume of cylinder in cubic feet.
P = Mean pressure in lb. per sq. in.
S = Mixture strength in B.Th.U. per cu. ft.

Then $$S = \frac{Q \times F}{M \times V}$$

and the " Tookey factor " $$= \frac{P}{S}$$

For the examples of gas engines worked out in detail in Tookey's paper, this factor varies from 1·83 to 2·43. This means that a mixture strength of 40 B.Th.U. per cu. ft. should give a mean effective pressure of just over 80 lb. per sq. in.

Now a volumetric efficiency of, say, 90 per cent. means that one-tenth of the charge consists of exhaust gases and that the remaining nine-tenths are fresh air and gas. Consequently a mixture strength of 40 B.Th.U. as computed by Tookey would represent a calorific value of $40 \times \frac{100}{90} = 44{\cdot}5$ B.Th.U. for the air and fuel alone. If there are x parts of air to one part of fuel and the lower calorific value of the gas is, say, 450 B.Th.U. per cu. ft., then

$$44{\cdot}5 = \frac{450}{x+1}$$

whence $$x = 9{\cdot}1$$

or there are 9·1 parts of air to 1 part of gas by volume.

This gives a method of computing the ratio of air to gas in an actual engine. Reference has already been made to this when discussing the construction of the Walker chart, and it may here be noted that this chart, or the analytical reasoning upon which it is based, may be used to compute the air-to-fuel ratio in any type of internal-combustion engine. The two points that are required are the energy at the end of compression and the energy at the end of combustion. Assuming that combustion is complete before adiabatic expansion commences, these points may be determined either graphically, by comparing an actual indicator card with the Walker chart, or analytically, by calculating the temperatures from the pressures and then using the specific-heat formulæ to determine the energies at the required points. The method of doing this for the constant-volume cycle, when a mixture of air and combustible is compressed, has been shown on p. 142, Chap. VIII.

Calculation of Air-to-Fuel Ratio from Composition of Fuel and Exhaust-Gas Analysis.—(*a*) *Gaseous Fuels* (*by Volume*).—A determination of the air used can be made if the composition of 1 cu. ft. of the fuel is known and if an *accurate* determination of the free oxygen and other constituents of the exhaust gases can be obtained. The first gives the minimum quantity of air theoretically required for complete combustion. The second gives the amount of excess air (if any) which has found its way into the exhaust. The sum of these two quantities is taken to represent the total quantity of air per cubic foot of gas.

The chemical equations for complete combustion of the various combustibles in fuel gas are as follows :

Hydrogen	$2H_2+O_2=2H_2O$
Marsh gas	$CH_4+2O_2=CO_2+2H_2O$
Ethylene	$C_2H_4+3O_2=2CO_2+2H_2O$
Carbon monoxide .	$2CO+O_2=2CO_2$

Hence, taking the chemical symbols to represent the proportions of the gases present by volume in 1 cubic foot of the fuel, the amount of air (*a*) in cubic feet for complete combustion is represented by the following equation :

$$a=\tfrac{100}{21}[\tfrac{1}{2}(H_2+CO)+2CH_4+3C_2H_4] \quad . \quad . \quad . \quad . \quad (1)$$

This is the theoretical or minimum amount of air required for complete combustion.

The volume of the products of combustion can be derived in a similar way from the above chemical formulæ. For complete combustion the exhaust should contain only CO_2, H_2O, and N_2. Any free oxygen found by analysis should indicate the presence of excess air. Using the above equations the volumes of these three exhaust constitutents are derived as follows :

$$\left.\begin{aligned} CO_2&=CH_4+2C_2H_4+CO \\ H_2O&=H_2+2CH_4+2C_2H_4+H_2O \\ N_2&=N_2+\frac{79a}{100} \end{aligned}\right\}=V \quad . \quad . \quad . \quad (2)$$

where the chemical symbols, as before, stand for the proportion of the gases by volume in 1 cubic foot of the fuel.

Let a=theoretical amount of air in cu. ft. as calculated by equation (1)
x=excess air over this amount in cu. ft.
V=total volume of exhaust per cu. ft. of gas burned, for theoretical amount of air, in cu. ft.
=sum of equations bracketed together under (2)
v=the same, less the amount of water formed.

The last value represents the doubtful quantity in this method. When the exhaust gases are analysed the steam should have condensed to water, which takes up a negligible amount of volume and may therefore be neglected. In order to ensure accuracy the exhaust gases must be cooled well below 100° C. in order to obtain this result.

The following relation then holds good :

$$\frac{x}{x+v}=\frac{O_2\times\frac{100}{21}}{100}$$

where O_2 is the *percentage* of oxygen by volume found in the exhaust gas by analysis.

From this relation the excess air

$$x=\frac{vO_2}{21-O_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

but $v=CH_4+CO+N_2+\frac{79}{100}a+2C_2H_4$
and a is given by equation (1).

So that, by combining these two, the general expression for excess air may be written:

$$x=O_2\frac{[1\cdot88H_2+2\cdot88CO+8\cdot52CH_4+13\cdot27C_2H_4+N_2]}{21-O_2}$$

where O_2 is the percentage of oxygen (by volume) found in the exhaust gas, and the other chemical symbols represent the proportions of these respective gases in the analysis of 1 cubic foot of original fuel.

As already pointed out, the exhaust gases require to be thoroughly dried and cooled to ensure success with this method, and great care must be taken in the use of the Orsat or other apparatus when determining the percentage of free oxygen (if any) in the exhaust products. A convenient check on the accuracy can be made by comparing the proportion of CO_2 obtained from calculation with that actually found by analysis of the exhaust.

Example.—The Report of the 1904 Committee of the Institution of Civil Engineers on the Efficiency of Internal-Combustion Engines[1] gives (on p. 273) the following percentage analysis of the gas used in engine "X."

		per cent. by volume.
Combustibles	Hydrogen	41·29
	Marsh gas	33·73
	Ethylene	—
	Carbon monoxide	7·13
	Unsaturated hydrocarbons	4·74
	Carbon dioxide	2·62
	Oxygen	0·27
	Nitrogen	10·22

Unsaturated hydrocarbons are aliphatic hydrocarbons which have the property of receiving hydrogen. They are frequently known as olfines with the general formula of C_nH_{2n}. The saturated hydrocarbons or paraffins have an extra hydrogen molecule thus: C_nH_{2n+2}. Marsh gas, CH_4, is saturated, whereas ethelyne, C_2H_4, is unsaturated. In the above analysis the unsaturated hydrocarbons may be taken as ethylene for the purpose of determining the theoretical amount of air.

Using formula (1), p. 150, the theoretical volume of air (a) is:

$$\begin{aligned} a&=\tfrac{100}{21}[\tfrac{1}{2}(H_2+CO)+2CH_4+3C_2H_4] \\ &=\tfrac{100}{21}[\tfrac{1}{2}(0\cdot4129+0\cdot0713)+2\times0\cdot3373+3\times0\cdot0474] \\ &=5\cdot04 \end{aligned}$$

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[1] Proc. Inst. C. E., vol. clxiii. (1905), p. 241. See also "The Gas, Petrol, and Oil Engine," by Sir Dugald Clerk and Burls, vol. ii. (1913), pp. 411 and 413.

The composition of the exhaust products of this engine are those used to determine the specific-heat relations of the " B.A. " results, p. 114. They are given by Sir Dugald Clerk[1] as follows :

	per cent. by volume.
Steam (assumed gaseous)	12·0
Carbon dioxide	5·0
Free oxygen	8·0
Nitrogen	75·0

The volume (v) of the exhaust products less the amount of water formed can be determined from equation (2), p. 150 :

$$v = CH_4 + 2C_2H_4 + CO + N_2 + \tfrac{79}{100}a$$
$$= 0·3373 + 2 \times 0·0474 + 0·0713 + 0·1022 + \tfrac{79}{100} \times 5·04$$
$$= 4·5856$$

Therefore the excess air

$$x = \frac{vO_2}{21 - O_2}$$

$$x = \frac{4·5856 \times 8·0}{21 - 8}$$
$$= 2·82$$

and the ratio of air to fuel by volume

$$a + x = 5·04 + 2·82$$
$$= 7·86$$

The general expression for excess air (p. 151) could be used instead of the above method. The term inside the square brackets represents v. In this case it becomes

H_2	$1·88 \times 0·4129 = 0·776$
CO	$2·88 \times 0·0713 = 0·206$
CH_4	$8·52 \times 0·3373 = 2·870$
C_2H_4	$13·27 \times 0·0474 = 0·628$
N_2	$1·00 \times 0·1022 = 0·102$
	4·582

against 4·5856 by the slightly more accurate method.

In the trials the actual volume of fresh air in the explosive charge was determined by means of direct anemometer measurement. For engine " X " the sample figures are given as 7·97 for the most economical load (approximately full load in this case) and 8·27 for half-load, as compared with 7·86 calculated by the above method. But the Committee are careful to point out that it is " questionable whether a fair sample of the exhaust gas was always obtained."[2] Moreover, it was only found possible to analyse one sample of the incoming gas, so that under the circumstances the above result may be considered as satisfactory.

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[1] " The Gas, Petrol, and Oil Engine," Clerk and Burls, vol. ii. (1913), p. 413.

[2] Proc. Inst. Civ. Eng., vol. clxiii. (1905), p. 273.

The volumetric efficiency of this engine was not specifically determined in these trials as they were primarily concerned with the separation of the various heat losses, but Clerk mentions [1] that the gas present in the charge before ignition was in the proportion of 1 volume of coal gas to 9·2 volumes of air and exhaust gases. As the lower calorific value of the gas is given as 574 B.Th.U. per cu. ft., the mixture strength would average

$$\frac{574}{9·2+1}=56·3 \text{ B.Th.U. per cu. ft.}$$

Producer gas, Mond gas, Blast-Furnace gas, and Coke-Oven gas are all suitable fuels for the gas engine which is properly designed to use them. Their chemical and physical properties differ widely, both in themselves and between each other, and a satisfactory working knowledge of their characteristics can only be obtained by studying the books and papers specially devoted to them. The following table is a digest of some of this literature and shows, in round figures, what some of the properties of such gases may be. It may serve as a guide to the kind of results which could be expected when a preliminary design is being considered, but any definite installation would require more accurate determinations of the gas actually to be used, and even then it will be found that considerable variations in composition and calorific value are bound to occur in practice, for which due allowance will have to be made in the design of the engine.

TABLE VIII.—REPRESENTATIVE VALUES OF GASEOUS FUELS.

—	Coal gas.	Producer gas.	Mond gas.	Coke-oven gas.	Blast-furnace gas.
Per cent. by volume of:					
H_2 . . .	48·0	17·0	23·0	50·0	3·0
CH_4 . . .	32·0	3·0	2·0	28·0	—
CO . . .	7·5	20·0	16·0	5·0	28·0
C_2H_4, etc. .	5·0	—	—	—	—
CO_2 . . .	1·0	7·0	13·0	3·0	6·0
O_2	0·5	—	—	1·0	—
N_2 . . .	6·0	53·0	46·0	13·0	63·0
Lower Cal. Val. . .	450·0	145·0	130·0	400·0	85·0
Theoretical air . . .	5·1	1·17	1·12	3·98	0·74
Actual ratio . . .	9·0	2·2	1·9	7·9	1·43
Mixture strength . . (B.Th.U. per cu. ft)	45·0	45·0	45·0	45·0	35·0
Theoretical/actual . .	1·765	1·88	1·70	1·975	1·93

The lower calorific value is given here in B.Th.U. per cubic foot. The theoretical air is calculated from the figures higher up in the column. The actual ratio includes the residual gases, that is to say, it takes the volumetric efficiency into account, and the mixture strength represents the L.C.V. divided by this figure plus one. It will be noted that this actual ratio compared with the theoretical air, as shown in the bottom line, need not vary very much with the different types of fuel.

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[1] "The Gas, Petrol, and Oil Engine," Clerk and Burls, vol. ii. (1913), p. 413.

(*b*) *Air-to-Fuel Ratio of Liquid Fuels* (*by Weight*).—Liquid (and solid) fuels contain carbon, hydrogen, and sulphur. As they are analysed by weight, it is necessary to derive the amount of air required for complete combustion of each constituent. This is done by considering the chemical equations from a molecular weight point of view, thus :

Hydrogen :

$$2H_2+O_2=2H_2O$$
$$4+32=36$$

or 1 lb. of H_2 requires $\frac{32}{4}=8$ lb. of O_2 for complete combustion. Similarly

Carbon :

$$C+O_2=CO_2$$
$$12+32=44$$

or 1 lb. of C requires $2\frac{2}{3}$ lb. of O_2.

Sulphur :

$$S+O_2=SO_2$$
$$32+32=64$$

or 1 lb. of S requires 1 lb. of O_2.

Now air contains 23 per cent. by weight of O_2. Hence the amount of air required is

Hydrogen : $8\times\frac{100}{23}=34{\cdot}8$ lb. of air
Carbon : $2\frac{2}{3}\times\frac{100}{23}=11{\cdot}6$,,
Sulphur : $1\times\frac{100}{23}=4{\cdot}35$,,

Hence 1 lb. of liquid fuel containing these combustibles requires

$$[34{\cdot}8\,H_2+11{\cdot}6\,C+4{\cdot}35\,S] \text{ lb. of air} \quad . \quad . \quad . \quad (1)$$

the chemical symbols standing, in this case, for the proportions by weight in 1 lb. of the fuel. This formula should be compared with that of Dulong quoted by H. Moore [1] and other authorities

$$a=0{\cdot}116[C+3(H-\tfrac{1}{8}O)] \text{ lbs.}$$

where C, H, and O are the percentages by weight of carbon, hydrogen, and oxygen, respectively, in the fuel. If there is any sulphur present, it may be accounted for by including $+\frac{4{\cdot}35}{11{\cdot}6}=0{\cdot}375S$ inside the square brackets. This formula assumes that any oxygen which may be present in the fuel is already combined with hydrogen in the form of water; but it is rather misleading, because the oxygen is found in analysis by difference, and this includes all possible errors of measurement. In most cases of hydrocarbons the oxygen content is very small, one or two per cent., and the term may safely be neglected, but the formula is necessary for alcohol.

The majority of engines in this group, which work on liquid fuel and compress an explosive mixture, use petrol, benzol, or alcohol in conjunction with some form of carburettor. Sir Dugald Clerk and G. A. Burls [2] discuss in detail the possibility of estimating the air-to-fuel ratio from the exhaust gas analysis, and conclude that "it does not appear practicable to infer the proportions of the mixture supplied by the carburettor to the cylinders from the analysis of the exhaust as usually conducted," but they show a method devised by Dr. Watson which estimates the ratio of air to petrol from the hydrogen content of the original fuel which gives better, though not completely reliable, results.

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[1] "Liquid Fuels for Internal-Combustion Engines" (1918), p. 174.

[2] "The Gas, Petrol, and Oil Engine" (1913), vol. ii., Clerk and Burls, p. 630.

Fortunately this type of engine works best on approximately the correct theoretical air, so that equation (1) or, if preferred, Dulong's formula, may be used for preliminary guidance.

Professor A. H. Gibson and H. Wright Baker carried out a comprehensive series of tests on petrol engines with the main object of determining the temperatures of cylinder heads and exhaust valves. They measured both their air and their petrol supplies directly and obtained the following results[1] on a single cylinder of a standard 30-H.P. water-cooled engine made by Armstrong-Siddeley for motor-car work. This cylinder was 3·5 in. in diameter and the stroke was 5·25 in. The compression ratio (*r*) could be varied from 4·01 to 5·19, and the petrol used was Pratt's No. 1 motor spirit, having a specific gravity of 0·724 at 15° C.

TABLE IX.—AIR-TO-PETROL RATIOS.

r.	Air-petrol ratios for							
	Maximum power.		Maximum efficiency.		Maximum valve temperatures.		Maximum cylinder-head temperatures.	
(R.P.M.)	1,500	1,800	1,500	1,800	1,500	1,800	1,500	1,800
4·01	13·5	14·0	14·7	15·7	14·5	15·5	—	—
4·26	14·0	14·3	14·7	15·8	14·5	15·4	—	—
4·48	14·3	14·5	14·8	16·0	14·7	15·5	—	—
4·77	14·3	15·0	15·0	15·8	14·8	15·6	—	15·1
5·07	14·3	15·3	15·3	16·2	15·0	15·7	—	15·3
5·19[2]	14·0	14·5	15·0	16·1	14·7	15·5	—	15·3

The average of all these values is 15, which is a figure very often quoted as the correct theoretical proportion of air to petrol by weight.

Light Fuel Oils.—The whole subject of fuels, gaseous and liquid, is of fundamental importance to the internal-combustion engineer, and he should possess sufficient knowledge of chemistry, and more especially of physical chemistry, to understand the significance of the books and articles specially devoted to the subject. Information about the physical and chemical properties of fuels in this book is not intended to take the place of such specialised books, but is supplied chiefly with the object of providing the student with data for examples.

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[1] "Exhaust-Valve and Cylinder-Head Temperatures in High-Speed Petrol Engines," Proc. Inst. Mech. Eng., Dec. 1923, p. 1066.

[2] These results are probably affected by the fact that at this compression ratio pinking took place except with rich mixtures.

The commoner forms of liquid fuel used in the class of engines which fall under the first group are known as petrol, paraffin, benzol, and alcohol.

Petrol is a mixture of aliphatic hydrocarbons, such as pentane (C_5H_{12}), hexane (C_6H_{14}), heptane (C_7H_{16}), octane (C_8H_{18}), and nonane (C_9H_{20}). It distils over from crude petroleum at the lowest temperatures (between 50° and 150° C.) and, as such, is distinguished from

Paraffin, which is distilled from the same source between 150° and 300° C. The border line between the two is not clearly marked, and both have approximately the same elementary composition.

Benzol is a coal-tar product and consists of 70 per cent. benzene (C_6H_6) and 30 per cent. toluene (C_7H_8). It may be used by itself or mixed with petrol. In the latter case, the best compression ratio lies between the values for petrol (5) and for benzol (7) shown in Table VII. on p. 148.

Alcohol (C_2H_6O) differs from the previous spirits in two respects. It is not a hydrocarbon since it contains oxygen as well, and it is obtained not from petroleum or coal, but from vegetable matter which, with proper organisation, can be made to produce an inexhaustible supply. For this reason, increasing attention is being paid to its possibilities for use in high-speed internal-combustion engines,[1] though it cannot be said that much practical progress is apparent at present. This is partly due to the fact that a considerably higher compression is desirable for alcohol than for petrol,[2] so that automobile engines designed for the latter are not suitable for alcohol, and partly to the fact that, owing to its intoxicating powers, the British Government require some denaturant to be added which will render it unsuitable for drinking. Moore points out [3] that, up to the present, such denaturants are unsuitable for use in engines, and that government restrictions on the manufacture of alcohol itself also introduce difficulties. There seems to be little doubt, however, that as the natural sources of power fuel, such as oil wells and coalfields, are gradually exhausted, the world will have to rely to an increasing extent on vegetable fuel which can be reproduced from year to year. Alcohol is at present the most promising of such fuels as far as oil engines are concerned.

Although petrol, paraffin, benzol, and to a lesser extent alcohol, as used in engines, are not pure chemical substances, it is possible to get approximate chemical formulæ which represent their elementary composition as far as their average physical properties are concerned.

The following table shows such average compositions with their corresponding chemical formulæ and their "equivalent" reactions with oxygen for complete combustion.

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[1] See, for instance, paper by W. Watson, "Benzol, Alcohol, and Mixtures of these Liquids with Petrol as Fuels for Internal-Combustion Engines," Proc. Inst. Automobile Engrs., vol. 9 (1914), p. 73.

[2] See Table VII. on p. 148.

[3] "Liquid Fuels for Internal-Combustion Engines" (1918), p. 58.

TABLE X.—APPROXIMATE FORMULÆ FOR LIGHT FUELS.

Fuel.	Average sp. gr. at 15° C.	Proportion by weight of			Corresponding chemical symbol.	Corresponding chemical reaction.
		C.	H_2.	O_2.		
Petrol .	0·74	85	15	—	C_8H_{18}	$2C_8H_{18}+25O_2=16CO_2+18H_2O$
Paraffin .	0·81	85	15	—	$C_{10}H_{22}$	$2C_{10}H_{22}+31O_2=20CO_2+22H_2O$
Benzol .	0·88	92	8	—	C_6H_6	$2C_6H_6+15O_2=12CO_2+6H_2O$
Alcohol .	0·80	52	13	35	C_2H_6O	$C_2H_6O+3O_2=2CO_2+3H_2O$

Such symbols, although very crude from a chemist's point of view, are useful to the engineer in estimating average values for the air to fuel ratio or for the calorific values of the fuels. By using the formulæ for the proportion of air to fuel by weight (p. 154) with the percentage values of C, H_2, and O_2 given in the above table, the ratios can be calculated by weight. For alcohol, which contains oxygen, Dulong's formula should be used. Again, the proportion of air to fuel (in vapour form) by volume can be obtained by applying the chemical reactions to the methods shown under gaseous fuels (p. 152). The lower calorific value of such fuels may be estimated from formulæ proposed by the late W. Inchley,[1] slightly modified to include the oxygen in alcohol. This formula is

$$7500\,C+29000\left(H-\frac{O}{8}\right)\text{ C.H.U. per lb.}$$

In this manner a very fair estimate may be made of the mixture strength per lb. of the fuel which is likely to be obtained when the volumetric efficiency is taken at 100 per cent.

TABLE XI.—CALCULATED PARTICULARS OF LIGHT FUELS.

Fuel.	Air-to-fuel ratio.				Lower calorific volume C.H.U. per lb.	Mixture strength C.H.U. per lb.
	By weight.	By volume.				
		Weakest.	Correct.	Strongest.		
Petrol . . .	15·0	91	60	19	10,830	67·7
Paraffin. . .	15·0	—	74	—	10,830	67·7
Benzol . . .	13·5	37	36	16	9,220	63·8
Alcohol. . .	9·0	25	14	7·4	6,400	64·0

The weakest and strongest air-to-fuel ratios (by volume) are obtained from figures given by Moore.[2] The mixture strength (100 per cent. volumetric efficiency) is the lower calorific value divided by the air-to-fuel ratio (by wt.) plus one.

The Formation of Mixtures as affected by Governing.—The speed of gas engines may be controlled either by varying the

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[1] "Theory of Heat Engines" (1920), p. 243.

[2] "Liquid Fuels" (1918): petrol, p. 51; alcohol, p. 56; benzene (taken as benzol), p. 54.

quality of the mixture, that is the ratio of air to fuel per working stroke, or by adjusting the *quantity* of the fixed proportion of air to fuel which enters every working stroke. Control, for the most part, is obtained by various mechanisms actuated by the sleeve of a governor which is driven by the engine and is, to that extent, automatic.

The speed of petrol engines used for mechanical transport is varied by means of a hand throttle or foot "accelerator," and no mechanical governor is fitted. The aim of the designer here is to obtain the most suitable mixture strength for each load, and in this sense the carburettor should act as an automatic governor.

Quality Governing of gas engines is achieved

(*a*) by throttling or restricting the gas supply whilst leaving the air inlet untouched,

(*b*) admitting the gas during a portion only of the suction stroke,

(*c*) cutting off the gas supply completely until the speed returns to normal, this method being sometimes known as *hit-and-miss governing*,

(*d*) by a combination of (*a*) and (*b*).

Since the greater proportion of the mixture is air, this variation in the gas supply will not greatly affect the total quantity admitted per stroke, so that quality governing results in practically the same compression pressure at all loads. The variation in mixture strength, however, alters the maximum explosion pressure and consequently the mean effective pressure of the cycle. If the mixture is allowed to get too weak, or in some cases if it gets too rich, combustion is not instantaneous, but may continue during the whole of the expansion stroke and even to the end of the exhaust stroke in a four-cycle engine. This may result in the next charge being fired before it enters the cylinder, and Clerk and Burls point out[1] that back-fires produced in this way are more often observed with quality-governed engines than with others.

Generally speaking, method (*b*) is preferable to method (*a*) in quality governing, if only for the reason that the gas may be admitted for the second half of the suction stroke and so produce a localised rich mixture round the ignition point.

Hit-and-miss governing is only a special form of quality governing because the governor mechanism operates on the fuel inlet valve, and not as a rule on the air inlet. It results in one or more cycles being carried out with air only, and consequently the next incoming charge of air and gas enters at a lower temperature, and is therefore more dense. It frequently happens, therefore, that the resulting explosion is greater than the normal.

The method, however, is in common use for small gas engines (up to 100 B.H.P.) because it is very economical and sensitive in its action.

Quantity Governing of gas engines may be achieved

(*a*) by throttling the mixture as it enters the cylinder,

(*b*) by varying the lift of a common admission valve,

(*c*) by cutting off the mixture at some point in the suction stroke

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[1] "The Gas, Petrol, and Oil Engine," vol. ii. (1913), p. 364.

These methods have much the same effect on the resulting working medium. As a greater or less mass is admitted the pressure at the beginning of the compression stroke, and therefore the compression pressure will vary much more than with pure quality governing.

This means that with a poor quality of fuel, like blast-furnace gas, care must be taken that the compression at light loads is sufficient to ensure an explosive mixture (say 120 lb. per sq. in. abs.). The mechanism of quantity governing on the whole is simpler and more reliable than that of pure quality governing, and the method is being used more and more for medium and large-sized gas engines.

In theory both methods have their good points, but in practice they are not quite so simple to carry out. Many makers have their own special methods of governing, which have been developed by long experience, and these frequently involve a combination of both quantity and quality governing. It may be mentioned here that the two-stroke cycle gas engine is governed by control of the gas-pump delivery, which works either by throttled admission or by means of a bye-pass between it and the main cylinder. If necessary, further control is obtained on a combined gas-and-air inlet valve to the cylinder proper.

The Control of Petrol Engines.—The main control of the speed of high-speed engines, using petrol or other light spirit which requires extraneous ignition, is carried out by throttling the incoming mixture of air and petrol vapour after it has passed through the carburettor. This throttling, in mechanically propelled vehicles, is controlled by the hand or the foot, and not by a rotating governor working off the engine crankshaft. The adjustment of mixture strength for different speeds is left to the carburettor, though in some cases hand adjustments for extra air or for altering the fuel supply are also fitted. In this respect the carburettor may be considered as part of the governing system. The general theory of the carburettor is explained later on in this chapter.

General Theory of the Formation of a Correct Mixture.—The two fundamental equations which may be used in this connection are the equation of flow

$$P_1-P_2=kw\frac{V_2^2-V_1^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and the equation of continuity

$$AVw=\text{a constant} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In these equations

P=pressure in lb. per sq. ft.
k=a coefficient
w=the density of the medium in lb. per cu. ft.
V=velocity of flow in ft. per sec.
A=area in sq. ft.

If the initial conditions are atmospheric and the medium is substantially at rest, then equation (1) becomes

$$P_1-P_2=kw\frac{V^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1a)$$

Let the following suffixes indicate various states of the process :

f for the fuel.
a for the air.
m for the mixture.
p for the engine piston.
c for the combustion chamber.
t for the throttle or valves.
i for the inlet valve.

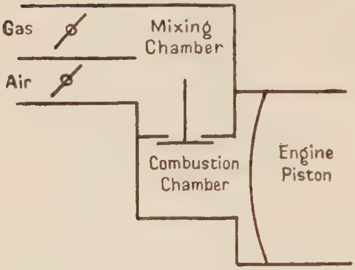

FIG. 48.—Gas Engine.

Then, in the case of the gas engine, Fig. 48, the pressure drops across the throttles are, for the air throttle :

$$P_a - P_m = k_t w_a \frac{V_a^2}{2g}$$

and for the gas throttle :

$$P_f - P_m = k_t w_f \frac{V_f^2}{2g}$$

If it is assumed that both the gas and the air are drawn into the system at atmospheric pressure, and that both the throttling effects are similar,

then $P_a = P_f$ and k_t is the same for both passages,

so that
$$w_a \frac{V_a^2}{2g} = w_f \frac{V_f^2}{2g}$$

or
$$\frac{\text{The density of the air}}{\text{The density of the gas}} = \frac{V_f^2}{V_a^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

This means that the velocities through the two passages are inversely proportional to the square roots of the densities.

Again, the equation of continuity shows that

$$A_p V_p w_c = A_i V_i w_i = A_a V_a w_a + A_f V_f w_f \quad . \quad . \quad . \quad (4)$$

In this relation V_p is taken as the mean piston speed of the engine.

If T_m is the temperature in the mixing chamber in °C. (abs.) and T_c the temperature in the combustion chamber during the suction stroke

$$w_c = \frac{T_m}{T_c} w_i$$

Hence the ratio of the effective area of the inlet valve to that of the piston

$$\frac{A_i}{A_p} = \frac{T_m}{T_c} \cdot \frac{V_p}{V_i} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4a)$$

If the velocity through the inlet valve is limited to a mean value of 110 ft. per sec. the volumetric efficiency will not suffer. It should not be allowed to exceed 150 ft. per sec. except in special cases. Equation (4*a*) then enables the effective area of the inlet valve to be calculated for any known piston speed and piston area.

Equation (4) also shows that

$$A_pV_pw_c=A_aV_aw_a+A_fV_fw_f$$

$$=A_aV_aw_a\left(1+\frac{A_fV_fw_f}{A_aV_aw_a}\right)$$

but $\frac{A_fV_fw_f}{A_aV_aw_a}=\frac{1}{Y}$ where Y is the air to fuel ratio by weight,

so that $A_pV_pw_c=A_aV_aw_a\left(1+\frac{1}{Y}\right)$

w_a is at atmospheric temperature T_a° C. (abs.)

so that $$w_c=\frac{T_a}{T_c}w_a$$

Then $$\frac{A_a}{A_p}=\frac{T_a}{T_c}\cdot\frac{V_p}{V_a}\left(\frac{Y}{Y+1}\right) \quad . \quad . \quad . \quad . \quad . \qquad (4b)$$

This equation may be used to determine the effective area of the air-throttle passage.

To determine the gas passage

$$Y=\frac{A_aV_aw_a}{A_fV_fw_f}$$

or $$A_f=\frac{A_a}{Y}\cdot\frac{V_a}{V_f}\cdot\frac{w_a}{w_f}$$

but $\frac{V_a}{V_f}=\sqrt{\frac{w_f}{w_a}}$ from equation (3)

so that $$A_f=\frac{A_a}{Y}\cdot\sqrt{\frac{w_f}{w_a}}\cdot\frac{w_a}{w_f}$$

$$=\frac{A_a}{Y}\cdot\sqrt{\frac{w_a}{w_f}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (5)$$

Example.—A gas engine with 9 in. dia. cylinder by 15 in. stroke runs at 160 r.p.m. It uses town gas whose density is 0·4 compared with air, and requires an air-to-fuel ratio of 5½ to 1. Determine the effective areas of the inlet valve and of the gas and the air passages.

The piston area $(A_p)=0{\cdot}785(\frac{3}{4})^2$
$=0{\cdot}442$ sq. ft.

The mean piston velocity $(V_p)=2\times\frac{15}{12}\times160=400$ ft. per min.
$=6{\cdot}66$ ft. per second

The velocity through the inlet valve (V_i) may be taken at 110 ft. per sec. If temperature of mixing chamber is 30° C. and of combustion chamber 100° C., then

$$w_c=\frac{273+30}{273+100}w_i=0{\cdot}81\ w_i$$

Hence by equation (4*a*),

$$A_i = 0{\cdot}442 \times 0{\cdot}81 \times \frac{6{\cdot}66}{110}$$

$$= 0{\cdot}0217 \text{ sq. ft.}$$

or effective area of inlet valve $= 3{\cdot}13$ sq. in.

Again, keeping the same mean velocity through the air throttle and assuming that the temperature of the incoming air is 15° C., by equation (4*b*)

$$A_a = 0{\cdot}442 \times \frac{273+15}{273+100} \times \frac{6{\cdot}66}{110}\left(\frac{5{\cdot}5}{5{\cdot}5+1}\right)$$

$$= 0{\cdot}01745 \text{ sq. ft.}$$

or area of air passage $= 2{\cdot}513$ sq. in.

Lastly, by equation (5)

$$A_f = \frac{0{\cdot}01745}{5{\cdot}5} \times \sqrt{\frac{1}{0{\cdot}4}}$$

$$= 0{\cdot}01745 \times \frac{1{\cdot}58}{5{\cdot}5} \text{ sq. ft.}$$

or area of gas passage $= 0{\cdot}721$ sq. in.

The Carburettor.—The carburettor, as has already been pointed out, is an apparatus used to obtain a suitable mixture for oil engines running on petrol and other volatile spirits. Its main function is to pulverise the petrol and to intermingle the finely divided spray with air to form, as nearly as possible, a homogeneous mixture of the correct proportions. As, however, the carburettor works by means of the suction of the engine, and not by an externally controlled feed pump, a further requirement in its design is some arrangement whereby the most suitable mixture is obtained whatever the speed of the engine may be. Flexibility of speed is one of the essential features of the automobile engine, and as the internal-combustion engine has a very poor starting torque, it is usually necessary to sacrifice economy by providing a specially rich mixture when the engine is running slow. This introduces a further complication in the proper design of the carburettor.

It will thus be seen that the name "carburettor," like so many scientific terms which have crept into practical engineering, is a complete misnomer for the apparatus as used for petrol engines. A carburettor, in the chemical sense, is an apparatus for adding carbon to gases, but that is the last thing which the carburettor of a petrol engine should do.

There are a great many designs of carburettors in use, and it is hardly possible to differentiate between the degree of success of the better known ones.[1] The majority of them work with some form of petrol jet which is placed inside an air nozzle, called a choke tube, the level of the petrol in the jet tube being maintained constant when at rest by a float in an adjacent float chamber (Fig. 49).

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[1] For details of various carburettors see "The Gas, Petrol, and Oil Engine," by Clerk and Burls (1913), vol. ii. chap. ix., or other standard works. For their practical design see "The Internal-Combustion Engine," by H. Ricardo (1923), vol. ii. pp. 119–130.

The same general theory as used above for a gas engine can be applied to the carburettor and the induction system of a petrol engine with one reservation, which was first pointed out by H. E. Wimperis.[1] In order to prevent spilling of the petrol when the vehicle is at rest, especially on a gradient, the level of the petrol in the jet must be kept by the float slightly below the tip of the nozzle. Consequently, a certain air velocity is necessary before any petrol will flow at all. Let this height be H ft. Then equation (3) when applied to the carburettor jet and choke tube becomes

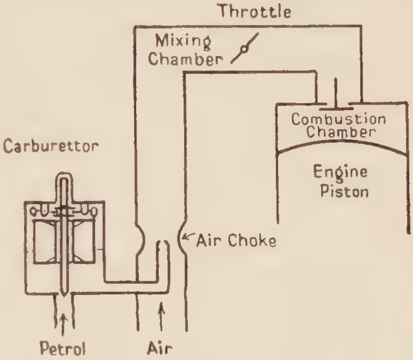

FIG. 49.—Principle of Carburettor for Petrol.

$$\frac{w_a}{w_f}=\frac{V_f^2+2gH}{V_a^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3a)$$

consequently the petrol does not leave the tip of the nozzle until the velocity of the air

$$(V_a) \text{ is greater than } \sqrt{2gH\frac{w_f}{w_a}}$$

Wimperis shows, in an example, that this function causes the petrol velocity to increase at a greater rate than the air velocity. Since the air velocity is controlled by the speed of the engine, this tends to give a richer mixture for a higher speed, unless some compensating device is included.

Furthermore, equation (3) is based on the assumption that the coefficient k is unity. This is not necessarily the case in practice. Equation (3*a*) then becomes

$$\frac{w_a}{w_f}=\frac{k_f}{k_a}\cdot\frac{V_f^2+2gH}{V_a^2}$$

where k_f is the coefficient of discharge of the petrol jet into a rapidly moving column of air, and k_a is the coefficient of discharge of the choke tube or air nozzle with a central tube inside it. The values of k_f and k_a can best be determined by direct experiment over the range of working of an actual carburettor. Petrol will only flow in this case when

$$V_a \text{ is greater than } \sqrt{2gH\frac{k_f w_f}{k_a w_a}}$$

Another point is well brought out by consideration of the general theory, and that is the value of the arrangement of a carburettor as an atomiser.

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[1] "The Internal-Combustion Engine," by H. E. Wimperis (1908), pp. 266 *et seq.*

Equation (3) shows that, neglecting the coefficients, the ratio of the velocity of the air in the choke tube to the velocity of the petrol leaving the petrol jet is inversely proportional to the square root of their respective densities. 1 cu. ft. of air at S.T.P. weighs 0·0807 lb. and at a temperature of 15° C. this reduces to

$$0{\cdot}0807 \times \frac{273}{273+15} = 0{\cdot}0765 \text{ lb.}$$

Petrol has an average specific gravity of 0·74 compared with water, hence 1 cu. ft. weighs $0{\cdot}74 \times 62{\cdot}3 = 46{\cdot}3$ lb.

Hence $$\frac{V_a}{V_f} = \sqrt{\frac{46{\cdot}3}{0{\cdot}0765}} = 25 \text{ to } 1 \text{ (approx.)}$$

or the petrol jet is entering a stream of air which is moving some twenty-five times as fast as itself. The pulverising action is therefore very considerable, if the proportions of the carburettor are properly adjusted.

Example.—Determine the effective area of the throat of the choke tube of a single-jet carburettor with a jet area of 2 square millimetres, which is required for a 20 B.H.P. petrol engine using 0·6 lb. of petrol per B.H.P. per hour.

The weight of petrol flowing per second

$$W_f = \frac{0{\cdot}6 \times 20}{3600} = \frac{1}{300} \text{ lb. per sec.}$$

Since 1 sq. centimetre $= 0{\cdot}155$ sq. in.

$$2 \text{ sq. millimetres} = \frac{0{\cdot}00155 \times 2}{144} \text{ sq. ft.}$$

but $$W_f = A_f V_f w_f$$

Taking the specific gravity of petrol when compared with water as 0·74, the density

$$w_f = 0{\cdot}74 \times 62{\cdot}3 = 46{\cdot}3 \text{ lb. per cu. ft.}$$

Hence the velocity of the petrol as it issues from the jet

$$V_f = \frac{1}{300} \times \frac{144}{0{\cdot}00155 \times 2} \times \frac{1}{46{\cdot}3} \left(\frac{\text{lb.}}{\text{sec.}} \cdot \frac{1}{\text{ft.}^2} \cdot \frac{\text{ft.}^3}{\text{lb.}} \right)$$
$$= 3{\cdot}35 \text{ ft. per sec.}$$

The velocity of the air in the choke tube has just been shown to be about 25 times this value, or

$$V_a = 3{\cdot}35 \times 25 = 83{\cdot}6 \text{ ft. per sec.}$$

The ratio of air to fuel is round about 15 to 1 by weight, hence the weight of the air

$$W_a = 15 \times \tfrac{1}{300} = \tfrac{1}{20} \text{ lb. per sec.}$$

The density of air will be very slightly affected by the small drop in pressure in the choke tube; neglecting this correction for the moment the value may be taken at 0·0765 lb. per cu. ft. at 15° C.

Then, since $W_a = A_a V_a w_a$

the effective area of the choke tube in square inches

$$a_a = \frac{W_a}{V_a \times w_a} \times 144$$

$$= \frac{144}{20 \times 83{\cdot}6 \times 0{\cdot}0765}$$

$$= 1{\cdot}127 \text{ sq. in.}$$

The drop in pressure may also be determined from equation (1*a*)

$$P_1 - P_2 = k_a w_a \frac{V_a^2}{2g}$$

The coefficient k_a will be of the order of 0·94 for a well-designed choke tube at a velocity of 100 ft. per sec.

Hence $$P_1 - P_2 = 0{\cdot}94 \times \frac{0{\cdot}0765 \times 83{\cdot}6^2}{64}$$

$$= 7{\cdot}86 \text{ lb. per sq. ft.}$$

or $$\frac{7{\cdot}86 \times 12}{62{\cdot}3} = 1{\cdot}5 \text{ in. of water.}$$

CHAPTER X

THE FORMATION AND FIRING OF MIXTURES IN WHICH AIR ONLY IS COMPRESSED, AND THE EXPANSION AND REMOVAL OF THE WORKING SUBSTANCE

It was pointed out at the beginning of the last chapter that the pressure and temperature of compression in internal-combustion engines in which air only is compressed is not limited by considerations of pre-ignition. Engines of this type inject their fuel at or near the end of the compression stroke into a mass of compressed and heated air. Ignition of the mixture falls broadly into three groups:

(*a*) Diesel engine type, which relies definitely on the temperature which the air has attained under compression;

(*b*) hot-bulb engine type, which uses a hot spot or surface in the combustion chamber;

(*c*) solid-injection engine type, which depends on a variety of causes outlined in Chapter III. p. 54.

The first group uses blast air, at a higher pressure than that which the air inside the combustion chamber has reached, to force the fuel oil into the cylinder. The second and third groups rely on some mechanical means, such as oil pumps or springs or a combination of both, to inject the oil. In all three cases combustion should commence as soon as the proportion of air to fuel has reached an explosive stage. In the case of the Diesel engine, this combustion takes place at approximately constant pressure, and the fuel is therefore definitely burning over the period during which the fuel valve remains open. The hot-bulb engine fires at constant volume, so that, in theory at least, the formation of the mixture and its ignition should both occupy the minimum amount of time. The solid-injection engine also fires at constant volume, but here, by suitably delaying the period of cut-off of the fuel valve, it is possible to obtain a certain amount of combustion at constant pressure as well, so that this type of engine may function on the dual-combustion cycle.

It cannot be too strongly emphasised that the Diesel engine functions in this respect in a fundamentally different way to the other two groups, and such names as " semi-Diesel " and " airless-injection Diesel " are extremely misleading when applied to hot-bulb or to solid-injection oil engines.

The problem, namely the formation of a homogeneous mixture of air and oil vapour in the most suitable proportions for its subsequent combustion in the most efficient manner, is the same, but each of these three groups of engines solves the problem in a different way.

Before considering these methods in more detail it should be pointed out that the various arrangements are equally suitable for engines which work on either the four-stroke or the two-stroke cycles.

The Formation and Combustion of the Working Substance in a Diesel Engine.—The fuel valve of a Diesel engine is shown in its simplest form in Fig. 50. It consists of a small central hole in a circular hardened steel disc which is let in flush with the wall of the combustion space. The entry to this nozzle orifice is conical in shape, and contains a conical-pointed needle valve which seats on to the back of the orifice when not in use. Surrounding the needle are a number of loosely packed perforated discs, with a grooved truncated cone piece fitting into the conical portion. There are many varieties of this arrangement of plates, but their object is to break up the fuel oil so as to expose as large a surface as possible to the blast air. The oil comes in under pressure through the passage shown on the left and the whole of the entry chamber is otherwise filled with blast air. Reference may be made to Fig. 18 (p. 45) to show the general setting of such a valve.

When the needle valve is drawn back, the oil already intermingled with the blast air is forced by the latter through the orifice and comes out into the combustion chamber in a finely divided spray cone. The valve lift, in practice, is very small, about two hundredths of an inch, and the diameter of the orifice varies from about ten to twenty hundredths of an inch, according to the horse-power of the cylinder.

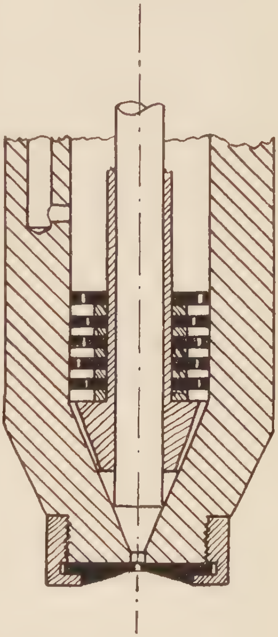

Fig. 50.—Elements of a Diesel-Engine Fuel Valve.

It will be apparent what a fairy godmother the blast air is to the Diesel engine. Not only does it ensure that the mixture of air and fuel is well intermingled as it enters the combustion chamber, but it also provides sufficient penetrating power to fill the space in a uniform manner. Moreover, the presence of blast air in the jet means that there should always be an explosive mixture formed from the moment the fuel valve opens, and the jet can be regulated to enter gradually so as to obtain combustion at approximately equal pressure. This gradual admission is essential for the Diesel engine, in which the air is compressed to some 500 lb. per sq. in.; otherwise the pressure would rise to dangerous (and unnecessary) heights.

Although the term "slow combustion" is sometimes applied to the Diesel-engine cycle, the actual time during which the fuel valve remains open is very short. Fig. 19, p. 47, shows that, for normal running, the fuel valve is open from 3 to 3*a*, or about one-eighth of a revolution in a four-stroke engine. For an engine running at 200 r.p.m. this represents a time interval of

$$\tfrac{1}{8}\times\tfrac{60}{200}\text{ second}$$

or approximately four-hundredths of a second. During this short period, the working substance has to be injected, vaporised, mixed, and burnt, so that even with the help of the blast air it is rather wonderful that the Diesel functions so well. As will be seen later, there are reasons for thinking that, in practice, "after-burning" occurs in most types of internal-combustion engines, and this would influence the shape of the expansion curve; this, however, does not prevent the ideal condition from being aimed at in design.

The Shape of the Combustion Chamber in Diesel Engines.—In a series of articles,[1] W. Stremme, of Budapest, discusses, amongst other matters, the effect of the shape of the combustion chamber on the formation and combustion of the mixture in Diesel engines. These articles are primarily concerned with the relative merits of short- and long-stroke Diesels, but it is claimed that the long stroke is superior, other things being equal, on account of the relatively larger combustion space which gives a lesser consumption due to a better mixture. He also claims that the turbulence effect, which makes for a more homogeneous mingling of air and oil vapour, is more pronounced in the long-stroke than in the short-stroke engine. The correspondence referred to in the same footnote is interesting, as it is claimed there that the objections to the shorter stroke may be overcome by making the piston top approximately cup shaped. Apart from increasing the relative size of the combustion space by such an arrangement, the theory underlying this idea is as follows: The usual Diesel-engine nozzle has a single orifice and the fuel blast air issues from it in the form of a wide cone which, in the open air at any rate, is fairly evenly filled with oil spray. If this type of spray is caused to enter a cup-shaped, or hemispherical, chamber it will penetrate to all parts of the piston top in approximately the same time.

The reply to this argument is also illuminating. Stremme says that Diesel himself, and most of the important firms after him, carried out experiments with this shape of piston. Such a form was acceptable on account of its inherent strength and also because of its smaller ratio of cooling surface to volume. Why was it not adopted? Stremme claims that one of the chief functions of the blast air, besides introducing the fuel, is to mix the vaporised oil quickly with the compressed air in the combustion chamber, by squirting against and rebounding from the hot piston face; this is particularly important with heavy tar oils. He goes on to point out that, with the hemispherical piston end the "rays" of air rebound along their own paths, instead of being scattered outwards in all directions, with consequent loss in efficiency. He quotes figures to show that an 80-H.P. single-cylinder Diesel used 0·447 lb. of oil per B.H.P. per hour with a hemi-spherical-ended piston, and 0·425 lb. with a flat-ended piston.

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[1] Z. V. D. I. vol. 60 (1916), pp. 561, 588, and vol. 61 (1917), p. 224; see also a letter by A. Balof, vol. 61 (1917), p. 224, and Stremme's reply, vol. 61 (1917), p. 226.

Another method he mentions is to use a fuel valve with a small mushroom head opening inwards into the combustion chamber in place of the more usual needle type of valve.[1] This tends to send the spray out laterally and gives better turbulence. A 500-B.H.P. 3-cylinder Diesel used 0·432 lb. of oil per B.H.P. per hour with cup-shaped pistons against 0·415 lb. with flat-ended pistons. In both trials, which were comparative, the load was 90 per cent. of the normal.

A similar result may be obtained in a different way. Instead of using a mushroom valve to distribute the spray sideways, the piston top may be slightly dished with a small pointed cone left in the centre, or a separate cone-pointed plug may be screwed into the centre to divert the spray.

It is doubtful whether special formation of the combustion chamber in Diesel engines has much permanent effect on their thermal efficiency, because the object which it seeks to achieve, namely a better distribution of oil spray, is generally obtained by the all-pervading air blast. The usual shape approximates to a disc or short cylinder, and the only care that is necessary is to see that this disc does not become too thin. For this reason a dished piston end is suitable for short-stroke Diesels.

The Junkers Diesel, which has opposed pistons, uses either two injection valves or one double one. The spray, which is introduced by air blast as usual, issues fan shaped instead of as a cone. These fans are arranged as near to the flat piston faces as possible so as to contain most of the air compressed in the combustion chamber between them. As the pistons move outwards, this air is drawn through the spray, and this undoubtedly makes for good intermingling of air and oil vapour. The compression ratio in the Junkers Diesel may be very high, as much as 16 in some cases, without loss in combustion efficiency.

The Formation and Combustion of the Working Substance in a Hot-bulb Engine.—The present types of hot-bulb engines are the direct descendants of the engine described in the patents of H. Akroyd-Stuart and C. R. Binney, which were taken out in 1890. (For description of this engine see Chapter III. p. 41.) At first sight this statement may seem contradictory, because the Hornsby-Akroyd engine of 1890 functioned like a gas engine in that the fuel was injected at the beginning of the suction stroke, and a gaseous mixture was submitted to compression. Moreover, the vaporisation took place in a vessel which was only connected to the compression chamber by a bottle neck. Actually the transition was quite natural. As the compression ratio was gradually increased in attempts to procure a higher thermal efficiency, it was found necessary to inject the fuel later and later in the cycle to prevent the danger of pre-ignition. The first step was to inject at the commencement of the compression stroke, instead of during the suction stroke (in a four-stroke engine), and this injection period crept up the compression stroke more and more until eventually the injection took place shortly before the inner dead centre. It is still possible to govern hot-bulb engines by retarding this injection period and using smaller quantities of oil as the load drops. The vaporiser also underwent transformation. It gradually assumed the shape of a bulb on the end of the cylinder. The compression space in the cylinder remained water-jacketed whilst the bulb, or part of it, was left unjacketed so as to attain a temperature sufficient to ensure ignition. The scavenging of this bulb has always been a difficulty, so that, other things being equal, the smaller it is the better for the creation of a good working medium. This applies more particularly to the two-stroke type.

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[1] There is an illustration of such a valve (amongst others) in Z. V. D. I. vol. 60 (1916), p. 1079, Fig. 13.

In modern hot-bulb engines, the spray is directed, not towards the on-coming piston but backwards, or upwards, into the hot bulb. This is good for ignition but bad for the formation of a homogeneous mixture, on account of the larger proportion of inert gases which remain in the bulb. On the other hand, the temperature inside the hot bulb is sufficiently high, under normal conditions of running, to vaporise the oil very effectively as soon as it has entered, and so relieve the sprayer of part of its function.

The hot-bulb engine does not use blast air to force the fuel oil into the engine, but relies on a mechanical pump and spring-seated valve which works, in principle, in a similar way to the one shown for the solid-injection engine (Chapter III. p. 55). The nozzle, however, lends itself to a large variety of designs, many of which employ a spiral-worm passage behind the orifice to impart a tangential as well as a radial velocity to the outcoming spray.

The Brons Oil Engine.—One interesting development of the hot-bulb principle, which is designed to overcome the difficulties of good intermingling referred to above, is known, on the Continent, as the Brons oil engine.

In this design, the size of the hot bulb is reduced to a minimum, and, moreover, its usual position is reversed. It projects into the compression space instead of opening out from it. The hot bulb has been inverted and becomes a cup. The fuel valve, which in this particular design must open inwards, opens into this cup and admits the crude oil about halfway up the compression stroke. A ring of small holes, between 1 and 2 mm. in diameter is drilled round the lower circumference of this cup as shown in Fig. 52.

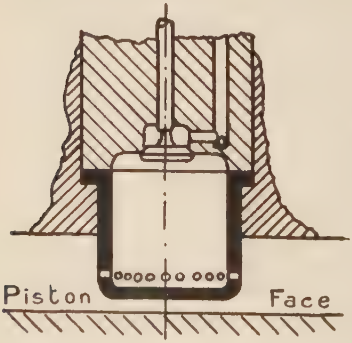

Fig. 51.—Hot Bulb of Brons Oil Engine.

Nearly all crude oils contain a certain proportion of volatile hydrocarbons, which form an explosive mixture with the air that has been forced into the cup through these holes by the incoming piston. This subsidiary explosion is sufficient to force the remaining portion of the fuel out through the holes which pulverise the oil into a finely-divided spray. By the time this action has taken place, the piston has reached the inner dead centre and the mixture is fired by the heat of compression, assisted, to an uncertain extent, by the heat of the cup.

Water Injection in Hot-Bulb Engines.—The general aspects of water injection into the cylinders of internal-combustion engines is discussed in the chapter on Temperature Effects (p. 193). Here it is only referred to in so far as it has been used for improving the combustion efficiency of hot-bulb engines.

The hot bulb has to be designed to function at all loads, and as it must of necessity be unjacketed, or partly unjacketed, it may get too hot under a prolonged heavy load. This has the double effect of causing a knock in the engine due to the mixture firing before the inner dead centre is reached, and, in extreme cases, of destroying the bulb. To avoid this, water injection into the bulb was at one time fairly common, and it still survives in certain types.[1]

Two methods were tried. A water drip which was introduced during the suction stroke, and which therefore did not require any water pump or other form of pressure apparatus, and a finely-divided spray introduced at the end of compression, either through a separate nozzle or through the oil-fuel nozzle itself. Under certain conditions, and particularly with heavy tar oils, it was found that the presence of the water actually reduced the consumption of the engine and tended to prevent carbon deposits. It is not clearly understood why this should be so, but it is assumed that at high temperatures (2000° C. and over) when water, or rather the steam formed from the water, dissociates into nascent hydrogen and oxygen. The nascent oxygen tends to combine with any carbon which would otherwise be deposited, whilst the nascent hydrogen, in certain fuels, reduces the heavy hydrocarbons to a lighter and more volatile series, and so improves combustion. This explanation must be considered as very problematical, as other reasons can be adduced to show that, in practice, water injection does not cause the improvements which have been claimed for it. The drawbacks to the use of water injection are discussed in Chapter XI. p. 193.

The Formation and Ignition of the Working Substance in a Solid-injection Oil Engine.—The formation of a homogeneous mixture of oil spray and air is a more difficult problem in the case of the solid-injection engine than in any other type of internal-combustion engine. The oil requires to be atomised sufficiently to ensure spontaneous and complete combustion, whilst, at the same time, it must have enough penetrating force to reach all parts of the compressed air in the combustion chamber. Failure in either of these two conditions leads to irregular, or heavy, consumptions, or, what is worse, to incomplete combustion with a smoky exhaust and a rapid deposition of carbon.

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[1] For a modern example see the "Dictionary of Applied Physics" (1922), vol. i. p. 332. For American practice see "Oil Engines," by Lacey H. Morrison (1919), chap. xxiii.

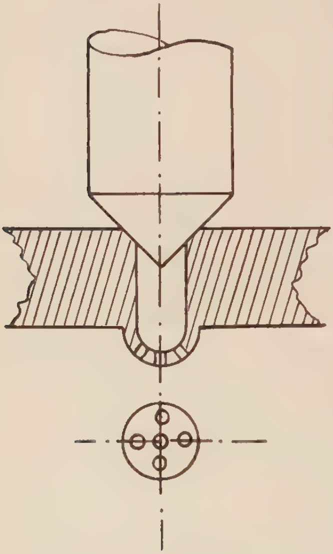

FIG. 52.—Elements of Solid-injection Engine Sprayer.

The nozzle usually consists of a round nose as shown in Fig. 52, through which are drilled a number of very fine holes between $\frac{1}{32}$ and $\frac{1}{64}$ of an inch in diameter. Behind these is a larger hole some $\frac{1}{8}$ of an inch in diameter, into which the point of the needle valve seats. The valve usually functions as explained in Chapter III. p. 55, and Fig. 23, but the holes are so fine that some form of oil strainer is necessary.[1]

Very little research work on the action of the solid-injection engine has been published, but in an instructive paper by Engr.-Commander Hawkes [2] it is pointed out (p. 60 of the paper) that "The number and size of the holes in a sprayer depend on the mean pressure in the fuel-injection system; on the viscosity of the fuel oil; on the mean indicated pressure aimed at; on the distance the jets have to penetrate into the combustion chamber, and also on the speed of the engine. The angle of the holes, *i.e.* the angle of the cone on which the holes are drilled, is dependent on the shape of the combustion chamber." This paper contains details of a number of tests carried out at the Admiralty Experimental Laboratory during the Great War. A single-cylinder four-stroke engine was used with a piston of aluminium alloy. The cylinder diameter was 14½ in., the stroke 15 in., and the normal rating 100 B.H.P. at 380 r.p.m. Various arrangements and numbers of nozzle holes were tried, but the best results were obtained with a sprayer containing five 0·016 in. holes, each about $\frac{1}{32}$ in. long, and an oil pressure of 4400 lb. per sq. in. When using shale oil (sp. gr. 0·87 at 60° F.) consumptions of 0·409 lb. per B.H.P. per hour were obtained over a six-hour run (p. 58 of the paper). The compression pressure in the engine was 380 lb. per sq. in.

The Function of the Holes.—There is still a good deal of obscurity as to what happens inside a "pipe orifice." The classic experiments and deductions of Osborne Reynolds, Lord Rayleigh, and Stanton and Pannell.[1] have shown that certain common laws of frictional resistance may be applied to both liquids and gases when flowing through pipes in which all disturbances, due to conditions of entry, have been carefully eliminated.

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[1] For a neat arrangement of solid-injection pump which works on the compression of the main cylinder, see "The Arschaouloff System" in *Engineering*, vol. cxviii., 1924, p. 29.

[2] "Some Experiments in Connection with the Injection and Combustion of Fuel Oil in Diesel Engines," by Engr.-Commander C. J. Hawkes, R.N. (ret.), Trans. North-East Coast Institution of Engineers and Shipbuilders, vol. xxxvii. (1920), pp. 37 *et seq.*

Again, the general behaviour of pure orifice flow is fairly well understood from many experiments which are to be found in standard works on hydraulics.

The author, however, has found from his researches, both with steam nozzles and with solid-injection oil nozzles, that neither of these sets of laws can be applied to what may be termed a pipe orifice, namely a comparatively short length of pipe, the flow in which is affected by conditions of entry. There is some disturbing influence at work the nature of which, at present, is not clearly understood.

The practical significance of this disturbance lies in the fact that when and where it occurs, discharges tend to become unstable. This results in irregular consumptions in the case of oil engines.

The oil-nozzle tests referred to above consisted in measuring the hydraulic consumptions of single-nozzle holes of varying diameter and length. The diameters were of the order of $\frac{1}{32}$ inch, while the entry channel consisted of a hole made by an $\frac{1}{8}$ in. twist drill.

Water pressure differences up to 1000 lb. per sq. in. were used, and irregular consumptions, amounting in some cases up to 10 per cent. variation, were found to occur in the region between 4*d* and 2*d* (see Fig. 53). A secondary disturbance, of the order of 5 per cent., was also observed at and below 1*d*. Beyond 4*d*, up to the pressure difference of 1000 lb. per sq. in., the discharge steadied itself and became regular. Fig. 53 is not intended to show the actual shape of the stream in the hole, but only to indicate the disturbed regions. The main disturbance, between 2*d* and 4*d*, varied within the pressure limits of 400 and 1000 lb. per sq. in.

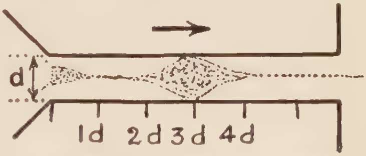

Fig. 53.—Oil-nozzle Experiments.

For the smallest hole tried (31 mm.), it was most pronounced at lower pressures. An intermediate hole (37 mm.) showed irregular discharges in this region over the whole of the pressure range, whilst in a larger hole (42 mm.) it was only present at 1000 lb. per sq. in. The secondary disturbance, below 1*d*, was always present, though the variation in discharge never exceeded half the value of the main disturbance.

The spray from these holes when discharging into the atmosphere looked like an elongated fox's brush. There were no signs of atomisation by the nozzle hole itself.

Irregular discharges were accompanied by an alteration in this shape. The single-cored jet broke up into a flat, fan-shaped jet with two boundary cores of water. Occasionally, though not always, at the very short lengths three or more cores were visible.

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[1] For a good summary of these see the article on "Friction," by T. E. Stanton in the "Dictionary of Applied Physics" (1922), vol. i. p. 356 *et seq.*

These experiments have not previously been published. They seem to the author to have only an indirect bearing on the problem of solid-injection oil-engine nozzles for two reasons. In practice the fluid is oil and not water, and the pressure differences are much greater, between 2000 and 10,000 lb. per sq. in. Moreover, the shape and dimensions of the entry channel will alter the condition of flow inside the pipe.

They are given here to show that there is something peculiar in the discharge of a pipe orifice. It is of interest to note, in this connection, that Commander Hawkes, in his tests referred to above, found that the best results were obtained after a small flat had been filed on each of his (16 mm.) holes, so that their length was reduced to about $\frac{1}{32}$ in. (31·25 mm. or $2d$).[1]

The function of the hole is not to atomise the fuel, but to direct the issuing jet to the proper quarter in the combustion chamber. There is very little, if any, latitude in the actual shape of the stable jet, but by drilling the nozzle holes at different angles, or arranging them unevenly round the nozzle nose, or using different sizes of holes, different shapes of the combined spray are possible.

Combustion Space of Solid-Injection Engine.—It is, however, in the author's opinion, much better to design the combustion space to suit such a spray, rather than to attempt to fill an existing shape. A suitable arrangement is shown in the diagram of the solid-injection engine in Chapter III. (Fig. 23) p. 55.

The solid-injection engine is the only one in which the fuel has to be atomised, vaporised, and mixed with the air after it has left the nozzle and entered the combustion chamber. Turbulence is therefore of considerable assistance, and this implies no pockets or sharp re-entrant angles in the space where the air may stagnate. On this hypothesis, the usual disc-shaped combustion chamber of the Diesel engine is not suitable for solid injection, as the resulting mixture is bound to be "stratified," or rather uneven in its richness.

In spite of such inherent difficulties, the solid-injection type of oil engine has already reached thermal efficiencies and actual fuel consumptions comparable with those of the Diesel engine; in conjunction with the Still system, it now provides the most economical heat engine which has yet been achieved.

The Expansion of the Working Substance.—The expansion of the working substance and its subsequent removal from the cylinder need not be subdivided into the two cases considered in the last two chapters.

The products of combustion, once firing has taken place, are similar in constitution for all types of internal-combustion engines. They consist of the following gases in varying proportions: carbon dioxide, steam, and nitrogen, with carbon monoxide if combustion is incomplete, and with oxygen if there is an excess of air.

Under faulty conditions of running, free carbon may be liberated either from the fuel oil or from the lubricating oil of the piston, in which case a carbon deposit is gradually formed inside the cylinder.

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[1] Page 53 of the paper.

The working stroke of all types of internal-combustion engines is therefore carried out by the expansion of a mixture of CO_2, H_2O

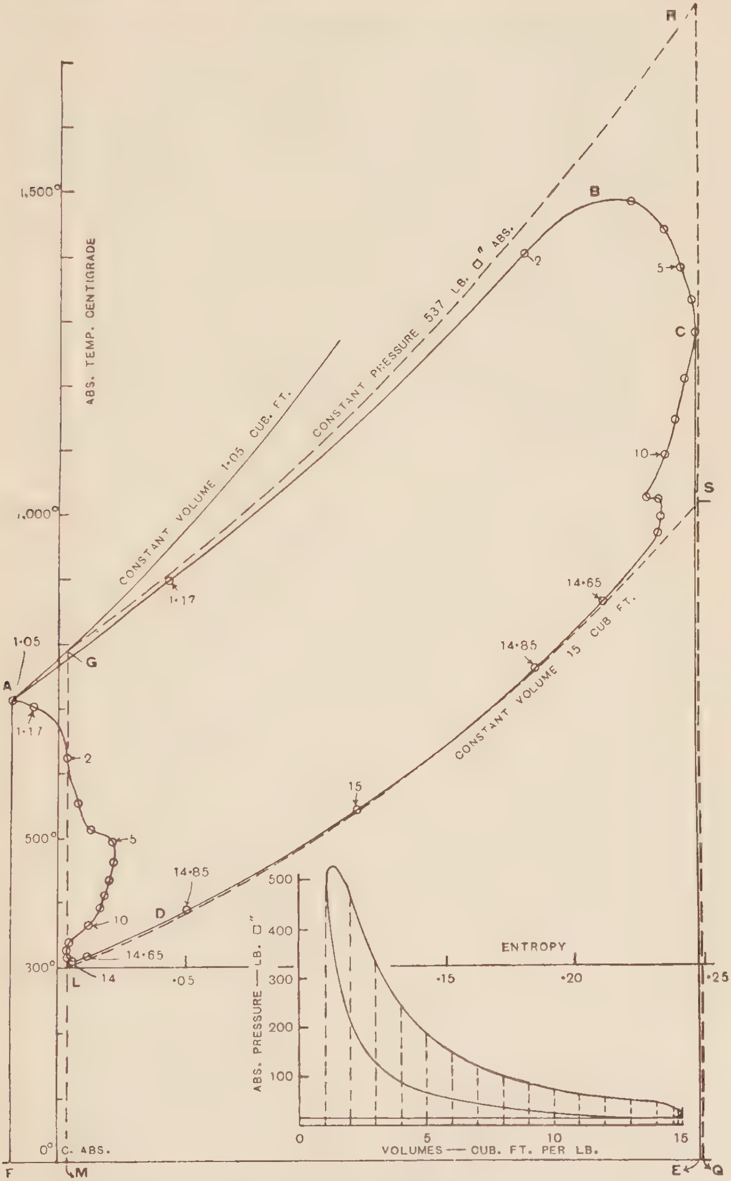

FIG. 54.—Application of Diesel Engine Card to Entropy Diagram.

and a number of diatomic gases. The internal energy of such a mixture is discussed in Chapter VII. The ideal condition postulates adiabatic expansion. If a reliable indicator card can be obtained, then by transferring it to an entropy chart, as described in Chapter VIII., it is possible to see at a glance how far the actual expansion curve deviates from the true adiabatic. Examples of such applications may be seen in Goudie's or in Alexander's papers referred to in the footnote on p. 129.

One of Alexander's examples is reproduced in Fig. 54, by kind permission of the Council of the Institution of Mechanical Engineers. It represents the case of a small Diesel engine tested by Professor Dalby,[1] which had a cylinder $6\frac{1}{2}$ inches diameter by $10\frac{5}{8}$ inches stroke. The compression ratio was 14·3. The card shows $10\frac{2}{3}$ H.P. developed at a speed of 259 r.p.m.

After-burning during the Expansion Stroke.—It will be noted that heat was being added to the mixture until the point C, halfway down the expansion curve, was reached. This implies after-burning, due to incomplete combustion of the fuel at the end of admission.

This particular example was chosen at random, but there is much other evidence available to show that combustion is seldom complete at the end of the so-called combustion portion of the ideal diagram This applies not only to Diesel and other oil engines but to gas and petrol engines as well.

Professor W. T. David, for instance, by a totally different method of reasoning,[2] shows that, under ideal conditions, only about 90 per cent. of the heat of combustion of the fuel of a gas engine has been converted into thermal energy by the time the maximum pressure is reached. In practice he finds that this "combustion factor" is only of the order of 70 per cent. He considers that the bulk of the after-burning is completed in the early stages of the expansion stroke, and is actually an advantage in that it reduces the ratio of maximum pressure to the mean effective pressure.

An analysis of nine continental Diesel engines in commercial operation by E. Weisshaar[3] claims to show the following average percentage of fuel oil burnt during the working stroke :

Percentage of working stroke travelled by piston.	Percentage of oil consumed.
up to 5	45·0
,, 10	72·5
,, 25	87·5
,, 50	93·5
,, 75	98·0
,, exhaust open	100·0

In four cases Weisshaar shows that combustion is not always quite complete when the exhaust valve opens. This investigation differs again from the two previously mentioned. It consisted in making a thorough heat analysis from complete trials (which are given in detail in the article) and estimating the flow of heat at different points of the working stroke from composite indicator cards direct.

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[1] Institution of Naval Architects, April 2, 1914.

[2] "The Missing Pressure in Gas Engines," W. T. David, Proc.I.M.E., October 24, 1924, p. 753.

[3] "Untersuchungen über den Vorlauf der Verbrennung im Dieselmotor," by E. Weisshaar, *Zeitschrift des Vereines deutscher Ingenieure*, vol. 60 (1916), p. 276.

There is a fourth method which may sometimes show after-burning, and that is an application of the actual indicator card to the Walker chart. Any gain in heat during the expansion stroke will show itself by altering the slope of the curve from the pure adiabatic.

An example is shown in Fig. 55. It is taken from Fig. 2, Plate I.

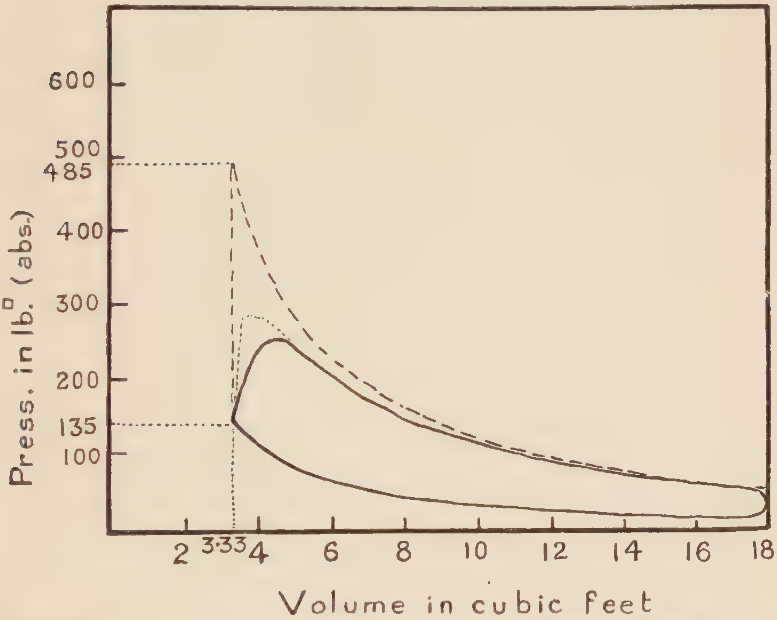

FIG. 55.—Comparison between Actual and Ideal Indicator Cards of a Gas Engine.

of Tookey's paper referred to on p. 142. The actual tests which included this card are given in Table I. on p. 9 of that paper, and the method of computing the ideal Walker chart card is shown on p. 140.

It will be noticed that there is anything but constant-volume combustion and that there must have been a considerable amount of after-burning. The chain line represents constant-volume combustion with adiabatic expansion according to Walker's chart. The intermediate dotted explosion line shows the improvement effected by Tookey in this particular engine.

CHAPTER XI

TEMPERATURE EFFECTS

The whole problem of the design of internal-combustion engines is very closely bound up with the effects which may be caused by the temperatures reached by the working substance. These effects fall roughly under the following headings:

Pre-ignition.
Excessive heat loss, detrimental to thermal efficiency.
Expansion of the metals used in construction.
Temperature stresses set up in these metals.
Reduction in the tensile strength of these metals at high temperatures.
Possible destruction of valves and valve-seatings, breech ends and piston crowns.
Growth of cast iron at high temperatures.
Evaporation of the lubricant of the piston.

All these effects have to be considered in a careful design, and their cumulative effect is, at present, the limiting factor in the size of the reciprocating internal-combustion engine, as well as one of the chief difficulties in the evolution of a satisfactory gas or oil turbine.

The question of pre-ignition has already been discussed, in part, in Chapter IX. pp. 147 *et seq.* A further reference will be made, below, to the use of water-injection in this connection.

When compression is increased with the object of improving the thermal efficiency, it can be shown that the rate of heat-loss per unit area may increase to an extent which will more than nullify this improvement.

Expansion of the metals is allowed for by the provision of sliding joints wherever they appear to be necessary, but the possible reduction in their strength at high temperatures is usually neglected.

Actual destruction of valves, valve-seatings, or other important surfaces, together with the destruction of the lubricant, is prevented by various artificial cooling devices such as water-jackets or air-cooling. If the stresses set up in the metal by temperature differences are still too high, then the only remedy is to reduce the strength of the mixture in order to keep the maximum temperatures down.

The possible physical growth of cast iron, which, in extreme cases, may amount to an increase of 36 per cent. in its volume,[1] is really a problem for the metallurgist. If suspected, it can be met, to some extent, by the use of special mixtures of metal together with more efficient methods of water-cooling.

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[1] Experiments on commercial irons by H. F. Rugan and H. C. Carpenter; quoted in "Cast Iron in the Light of Recent Research," by W. H. Hatfield (1918), p. 129.

The Transmission of Heat. Thermal Conductivity.—The thermal conductivity (k) of a substance is defined as the amount of heat which will flow through unit thickness, per unit area, per unit time, for one degree difference between the two faces.

In C.G.S. units the thermal conductivity equals the number of calories per second that would flow through every square centimetre of surface of a wall 1 cm. in thickness and with a difference of 1° C. between its faces.

The British units are usually taken as the B.Th.U. per hour, per square foot, for one inch thickness, per 1° F. difference.

The British unit, as defined above, equals 2903 C.G.S. units.

This relation may be put in another form:

$$H=\frac{kAZ(t_1-t_2)}{X} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

where H=amount of heat flowing
k=thermal conductivity
A=surface area
Z=time
(t_1-t_2)=temperature drop through wall
X=thickness of wall.

The relation $\frac{t_1-t_2}{X}$ is known as the temperature gradient, and represents the drop in temperature per unit thickness of the medium.

The thermal conductivity of cast iron and steel is not known very accurately. It is known to decrease very slightly with a rise in temperature, but actual values differ as given by various authorities.

This fact is not surprising when the variation in the composition of such metals is considered. A table of values by various authorities, given by Poynting and Thomson,[1] shows variations between 600 and 507 at 32° F. in British units, with corrections for increasing temperature of the order of $(1-0{\cdot}001t)$. Callendar and Nicolson,[2] by means of a series of specially devised experiments, deduced the values given in first table on p. 180 for cast iron.

Corresponding values for steel and for aluminium alloys are given in the article on "Heat Conduction" in the first volume of the "Dictionary of Applied Physics."[3] Steel (1 per cent. C) at 100° C. (212° F.) has a thermal conductivity of 311 in British units or 0·107 in C.G.S. units. The value does not seem to be more accurate than to within 5 per cent., so that for practical purposes, such as the steel liners of petrol engines, values similar to those for cast iron may be used without serious error.

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[1] "Heat," by Poynting and Thomson, 3rd ed. (1908), p. 99. Quoted by Professor Dalby in his paper on "Heat Transmission," Proc. Inst. M.E. (1909), p. 941.

[2] "Laws of Condensation of Steam," by H. L. Callendar and J. T. Nicolson, Proc. Inst. C.E., vol. cxxxi. (1897), p. 159.

[3] D.A.P., vol. i. (1922), p. 444, artcle by F. H. Schofield.

TABLE XII.—CONDUCTIVITY OF CAST IRON.

Temperature, °F.	Conductivity, *k*. B.Th.U. per sq. ft. 1 in. thick, per 1° F. difference, per hour.	*k* in C.G.S. units.
100	333·0	0·1150
150	328·8	0·1134
200	324·0	0·1117
250	318·6	0·1100
300	312·6	0·1080
350	306·0	0·1055

The values for aluminium alloys were specifically determined for aeroplane engines[1] by E. Griffiths, in 1917. Contrary to cast iron and pure metals generally, alloys show a slight tendency to rise with increasing temperatures.

TABLE XIII.—CONDUCTIVITY OF ALUMINIUM ALLOYS.

Composition.	Temperature.		Conductivity (*k*).	
	°F.	°C.	British units.	C.G.S. units.
1·5 Mg, 2 Ni 8 Cu, 88·5 Al	212	100	1,130	0·39
	572	300	1,160	0·40
1 Mn, 8 Cu 91 Al	212	100	1,045	0·36
	572	300	1,130	0·39
1 Ni, 8 Cu 91 Al	212	100	1,190	0·41
	572	300	1,218	0·42
2 Fe, 8 Cu 90 Al	212	100	1,102	0·38
	572	300	1,130	0·39

These values are about $3\frac{1}{2}$ times the thermal conductivity of cast iron and of steel, which is fortunate for the designer, because it enables him to make an aluminium piston crown, for instance, $3\frac{1}{2}$ times as thick without restricting the heat flow.

Example.—The 5th report of the Gaseous Explosions Committee of the British Association (1912) states (p. 193) that "the temperature gradient necessary to sustain the flow of heat from the inside of a combustion chamber to the external water is not likely to exceed 50° C. per inch." What average rate of heat flow does this represent?

50° C.=90° F. and *k* for C.I. at 100° F.=333 British units (from table XII.).

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[1] Advisory Committee for Aeronautics, Light Alloys Sub-committee Report, No. 7, November 1917. Quoted in D.A.P. article, *supra*.

Hence by formula (1) the heat flow

$$H=\frac{kAZ(t_1-t_2)}{X}$$

$$=\frac{333\times 1\text{ sq. ft.}\times 1\text{ hr.}\times 90^\circ\text{ F.}}{1\text{ in.}}$$

$$=29970\text{ B.Th.U. per hour, or}$$
$$500\text{ B.Th.U. per minute}$$

for a liner 1 inch in thickness.

Note.—The maximum temperature of the jacket water will not exceed the atmospheric boiling point of water, unless the jackets are under pressure, so that the maximum average temperature of the inside of a cylinder liner 1 in. thick will never be more than 150° C., when equilibrium of heat flow is established.

Conditions of Heat Transfer in an Internal-Combustion Engine.—A distinction should be made between the transfer of heat through a substance, determined by means of the thermal conductivity of that substance, and the transfer of heat across the boundary formed by two different substances. In an internal-combustion engine not only has heat to be conducted through the metal walls of chamber, but transfer of heat takes place between a gas and a metal, and between a metal and a liquid.

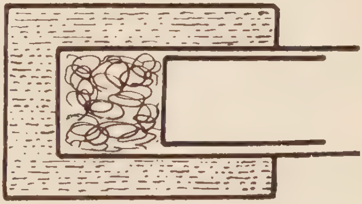

FIG. 56.—Heat Transmission in I.C. Engine.

The problem of heat transmission as applied to the internal-combustion engine is shown in its simplest form in Fig. 56. A cylindrical chamber, of varying length, contains hot gases in a high degree of turbulence. The temperature of these gases undergoes a cyclic variation between the temperature of suction, say 100° C., and the maximum temperature reached after combustion. This may be 2000° C. or even more. The range of this variation depends chiefly on the speed of the engine, the strength of the mixture, and whether the engine is functioning on a two-stroke or on a four-stroke cycle. It also depends, though to a lesser extent, on the type of engine and the method of introduction and exhaust of the working substance.

The best mineral cylinder lubricating oils evaporate rapidly if their temperature is allowed to exceed 315° C., so that the average temperature of the cylindrical sides of both the liner and the piston must be kept at or below, say, 300° C. This is not a difficult matter for the water-jacketed portions, as has already been seen, but the sides or trunk of the piston are in direct metallic contact with the uncooled top. The central portion of this crown may easily reach such a temperature that its rim either exceeds 300° C. or, if kept below that figure by the effect of the water jacket, the temperature stresses set up in this disc become excessive. The result is to crack the piston top. It may be mentioned here that the water jacket is extended along the cylinder liner chiefly with the object of cooling the piston rim.[1]

There are, therefore, two, or possibly three, conditions of heat transfer to be considered. There is first the case of the water-jacketed portion. Heat transfer takes place from hot gas to metal, then through the metal, and lastly from metal to water. The transfer from gas to metal is complicated by the presence, over a portion of the surface at any rate, of a thin film of lubricating oil.

The second case is that of the uncooled piston crown. Heat passes from a gas to a circular surface, the rim of which must be kept below 300° C., whilst the central portion may be much hotter. There is, therefore, a temperature gradient inside the metal between the centre and the rim. From the rim the heat flows down the cylindrical piston trunk, across the lubricant, into the liner walls and so out into the cooling water.

The third case of heat transfer really forms part of the second. It refers to the portion of the heat which is radiated out into the air from the inside of the piston top and sides. There is evidence which goes to show that such a radiation effect plays only a minor part in the total heat transfer from the piston.

Theory of Heat Transmission by Analogy with Boilers.—In one of the most valuable papers ever written on the subject,[2] Professor W. E. Dalby collects and gives a digest of all the important papers on Heat Transmission which had been written up to the year 1909. The subject-matter is dealt with from the point of view of heat transmission in steam boilers, but, with slight modification, it may be used in connection with the transfer of heat in internal-combustion engines.

Consider the general case of a metal wall separating two media at widely differing temperatures. If the heat transmission is much more rapid between one of these media and the wall than between the wall and the other medium, then the wall itself will be maintained at approximately the temperature of the medium with rapid heat transfer. In the case of a steam boiler (as well as that of an internal-combustion engine), Professor Dalby shows in his paper that there is an inert layer or film of gas which clings to the metal surface and which has a very poor thermal conductivity. The presence of this gas film, in the case of a boiler, is the predominating feature in the problem of heat transmission. Heat transfer between a hot gas and a surrounding metal wall is very sluggish on account of it. There is also a film of water clinging to the other side of the wall, but in this case the thermal conductivity is not greatly affected.

These facts are brought out in a diagram (Fig. 57) which is taken from Fig. 3 in Professor Dalby's paper, by kind permission of the Council of the Institution of Mechanical Engineers.

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[1] This was first pointed out by H. E. Wimperis in the discussion on "Cyclical Changes in a Gas-Engine Cylinder," by E. G. Coker and W. A. Scoble, Proc. Inst. C.E., vol. cxcvi. (1913–14), p. 57.

[2] "Heat Transmission," by W. E. Dalby. Proc. I.M.E. (1909), pp. 921–986.

In describing this figure, Professor Dalby says, "Starting from a point *a* in the interior of the hot gas, A*a* represents the temperature, and this is shown as gradually falling to the value B*b* at the surface of the gas film. The resistance of the film is extremely great, consequently the temperature falls to C*c* in passing through it, the transfer of heat having been effected by the temperature head B*b*—C*c*. There is probably a further drop of temperature head, represented by *cd*, which is required to force this flow across the surface where the gas film is in contact with the metal, corresponding with the potential difference at a joint in an electric circuit. The temperature head required to cause the flow through the metal itself is C*d*—E*e* with a further fall of head *ef* at the surface of contact between the metal and the water film. The fall of temperature head through the water film is E*f*—G*g*, until finally the temperature has fallen to that of the water, and the heat transmission through the mass of the water is continued by movement of the water particles."

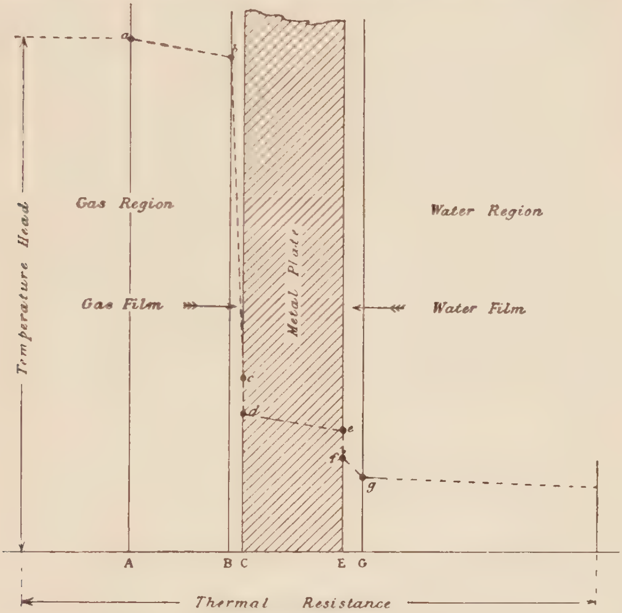

FIG. 57 —Heat Transfer through a Metal Wall.

The figure is not drawn to scale, since in the case of a boiler it is estimated that 98 per cent. of the total temperature head is absorbed in the gas film. Experiments are cited (p. 933 of Dalby's paper) to show that the thickness of such a film is of the order of $\frac{1}{40}$ inch.

The late Professor B. Hopkinson[1] placed patches of tin-foil on the inner surface of the combustion chamber of a gas-engine cylinder

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[1] Quoted in D.A.P., vol. i. (1922), p. 294.

$11\frac{1}{2}$ in. in diameter, and found these were quite unaffected by the successive explosions, although the heat flow here was a maximum and the melting point of the tin about 230° C. In another experiment with a closed explosion vessel,[1] Hopkinson also found that a layer of brown paper $\frac{8}{1000}$ inch thick was not even charred when a mixture of gas and air was exploded inside. Both these experiments go to prove the existence of a similar inert gas film in the case of an internal-combustion engine.

In practice this film is probably complicated by two factors, the possible presence of lubricating oil and any carbon deposits which may have formed. Both these substances are very poor conductors in comparison with metal.

The thermal conductivity of mineral oil is of the order of 0·0004 and of carbon (charcoal) about 0·000175 in C.G.S. units as compared with 0·11 for cast iron. The thermal conductivity of the carbon deposits would be of the same order as charcoal, namely, 0·000175 in C.G.S. units.[2] The presence of a lubricating-oil film would therefore tend to increase the heat loss between the gases and the metal wall, whilst the carbon deposit would not materially affect the conditions of heat transfer by conduction. As will be seen later, it plays an important part in radiation effects which are not, at present, being taken into account. Osborne Reynolds[3] showed that the total heat transmission in steam boilers may be expressed by the formula

$$H=(A+BSc)(t_1-t_2)$$

where H=amount of heat transmitted per unit of surface in unit time;

S=density of gas;

c=velocity of gas across the heating surface;

t_1=temperature of the gas;

t_2=average temperature of the heating surface;

and A and B are constants.

The first term represents what Osborne Reynolds called the natural internal diffusion of a gas when at rest. It depends only on the nature of the substance. The second term represents turbulence. Its magnitude is proportional to the density of the gas and to the velocity with which it flows past the surface.

It follows from this law, which has been fully corroborated experimentally, that the rate of heat transfer in an internal-combustion engine will increase with the density, in other words, with the compression, and also with the degree of turbulence present. The increase will not be directly proportional to the density (or to the degree of turbulence), because of the first factor represented by

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[1] Phil. Trans. Roy. Soc., A., vol. 211, p. 375. Quoted in 5th Report of the Gaseous Explosions Committee of the British Association (1912), p. 195.

[2] D.A.P., vol. i. (1922), article on Heat Conduction, p. 433.

[3] "On the Extent and Action of the Heating Surface for Steam Boilers," by Osborne Reynolds, Proc. Lit. and Phil. Soc. of Manchester, 1874, reprinted in "The Steam Engine," by J. Perry (1900), p. 594.

$A(t_1-t_2)$; but the fact remains that the rate of heat loss does increase with high ratios of compression, and a point may be reached where this increase in loss may more than counterbalance the improvement in thermal efficiency deduced from theoretical considerations.[1]

Again, a high degree of turbulence, which is essential in solid injection engines, for instance, and always present in Diesel engines owing to the air-blast, will also increase the rate of heat transfer according to the Osborne Reynolds law. It does so in two respects, by scouring the inert gas film and so diminishing very considerably the poorness of its conductivity, and by increasing the convection currents inside the gas itself.

Experimental evidence of this is found in the work of Professor A. H. Gibson and W. J. Walker, who tested a gas engine, *inter alia*, at different speeds, and state that "the rate of transmission of heat through the cooling surfaces is evidently much greater at the highest speed. An examination of the indicator diagrams, moreover, shows that the maximum pressure and temperature attained in the cylinder is approximately 6 per cent. greater at 150 than at 250 revolutions per minute, so that this increased rate of transmission is obtained in spite of a lower gas temperature. The reason is apparently to be found in the fact that the greater turbulence of the working fluid at the higher rates of speed increases its effective conductivity to an extent which more than counterbalances the effects of a smaller temperature difference and a shortened time of contact. Other things being equal, a 6 per cent. increase in the temperature of the gases would probably increase the heat transmitted by conduction and radiation by some 15 per cent., so that it may be taken approximately that the effective conductivity is increased in the same ratio as the speed of the engine."[2]

Radiation Effects.—So far, the question of heat transmission has been considered chiefly by the effects of conduction and convection, but as Sir Dugald Clerk points out,[3] after fitting a cylinder with a stout observation plug of glass, "while the engine is at work, a continuous glare of white light is observed; a look into the interior of a boiler furnace gives a good notion of the flame filling the cylinder of a gas engine."

The maximum temperatures reached in a small gas engine have been actually measured by Professor E. G. Coker and W. A. Scoble.[4] The engine had a 7 in. diameter cylinder with a 15 in. stroke, and developed 12 B.H.P. when running at 240 r.p.m. The ratio of compression was 5·8 to 1. In these experiments, specially constructed thermo-couples of platinum-iridium and platinum-rhodium alloys were inserted into the combustion chamber from above and suitably protected from the violence of the explosion. The results show that, with a 12 per cent. ratio of gas to cylinder contents, the highest temperature of the explosive charge reached 1836° C. With 13 per cent. mixture, this temperature rose to 1947° C., whilst with a 15 per cent. mixture, 2249° C. was recorded. The average lower calorific value of the gas used was 480 B.Th.U. per cu. ft.

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[1] For density effects see 5th Report of the Gaseous Explosions Committee of the British Association (1912), p. 197. There is a summary of this report in D.A.P., vol. i. (1922), p. 295.

[2] "The Distribution of Heat in the Cylinder of a Gas Engine," by A. H. Gibson and W. J. Walker, Proc. Inst. Mech. Eng. (May 1915), p. 376.

[3] D.A.P., vol. i. (1922), p. 293.

[4] "Cyclical Changes of Temperature in a Gas-Engine Cylinder," Proc. Inst. Civ. Eng., vol. cxcvi. (1913–14), p. 1.

Such temperatures represent a white-hot flame which would distribute most of its energy by radiation whilst in that condition. It is possible to form an estimate of the temperature in any particular portion of the heat cycle, as represented by an indicator card, if the pressure and the corresponding volume can be determined. The gas constant may be deduced from the ratio of the specific heats of the working substance

$$R=K_p-K_v$$

and the corresponding temperature obtained from the relation $\frac{PV}{T}=R$.

Alternatively, if two points are chosen which lie on either side of the ignition, and the products of combustion are x times the specific volume of the original working substance, then

$$\frac{P_1V_1}{T_1}=x\frac{P_2V_2}{T_2}$$

but this method has the drawback that it is a difficult matter to determine the temperature T (in absolute units) at any one point of the cycle unless special forms of thermo-couples are fitted.

It is, however, clearly established that the phenomenon of heat transmission by radiation must be present in all types of modern internal-combustion engines during some part of their cycle, and it remains to be seen how far such radiation effects can be reduced.

The Nature of Radiation.—The modern conception of radiant energy is very clearly summarised in Sir Oliver Lodge's book on "Atoms and Rays," which was first published in 1924. Put very briefly, it is now considered to be caused when electrons which normally revolve round the nuclei of atoms in a series of orbits, are moved from one orbit to another. In the words of Sir Oliver Lodge (p. 141, *supra*): "Some energy must be supplied to raise the particle from an inner orbit to any outer one. And, conversely, if the particle drops from any outer orbit to any inner one, some energy must be emitted or radiated away." In the latter case, the energy is transmitted through the ether in the form of waves, but the lengths of the waves may extend over a wide range. The shorter wave lengths give the sensation of light, whilst longer ones can be detected in the form of heat. The transmission of heat by radiation is akin to the transmission of light, and obeys similar laws of reflection, refraction, and absorption. It differs from conduction or from convection, in that the intermediate medium (the ether) is not itself affected.

Just as a highly polished surface reflects light, whilst a dull black one absorbs it, so is radiant heat reflected from a bright surface and absorbed by a black one. Roughly speaking, a metal surface coated with lamp-black or carbon will absorb about ten times as much radiant heat as the metal itself without any coating.[1]

This accounts, in part, for a fact which has long been recognised, namely, that an internal-combustion engine with a brightly polished interior will give a higher mean effective pressure than one with dull black sides and piston crown. In the former case, radiated heat is reflected and returned to the gas, adding to the quota of heat turned into work. In the latter case the radiant heat is practically all absorbed by the carbon deposit, and transferred by conduction through the cylinder walls to the cooling medium.

Radiation effects are always present during incandescence. A coke fire, which glows a bright cherry red, say 800° C., emits heat by radiation to the surrounding air. But the degree of radiation depends on the emissive properties of the incandescent substance. A black body, defined by Kirchhoff[2] as one which has the property of allowing all incident rays to enter without surface reflection, and not allowing them to leave again, obeys the Stefan-Boltzmann law

$$H=\sigma(T_1^4-T_2^4)$$

where H=amount of energy radiated in the form of heat ;
σ=a numerical constant determined by experiment ;
T_1=the temperature of the surface of the radiating substance in °C. abs. ;
T_2=the temperature of the surroundings receiving the radiation in °C. abs.

Lamp-black, in practice, nearly approaches the ideal black body, and its emissive properties are taken as 100 for a standard. The value of the numerical constant σ in the Stefan-Boltzmann law for an ideal black body is $5{\cdot}32\times10^{-5}$ ergs per sq. cm. per sec.[3] Since 1 calorie equals $4{\cdot}18\times10^{7}$ ergs,

$$\sigma=\frac{5{\cdot}32\times10^{-5}}{4{\cdot}18\times10^{7}}=1{\cdot}27\times10^{-12} \text{ cal. per sq. cm. per sec.}$$

This is shown by Professor Dalby[4] to reduce to 16×10^{-10} B.Th.U. per sq. ft. per hour, where $(T_1^4-T_2^4)$ is in degrees Fahrenheit.

These figures fix the maximum upper limit of radiation, and may be taken to apply to a steady flame composed of masses of incandescent carbon particles, a condition of affairs which is nearly approached in the fire-box of a locomotive. Inside the combustion chamber of an internal-combustion engine the conditions are more complex. The flame is not steady but intermittent, and moreover its nature is changing during each ignition. That solid particles of carbon are present is evident from the carbon deposits which may form, but other products of combustion, particularly CO_2 and H_2O, are also good emitters of radiation. Air, nitrogen, oxygen, and hydrogen, on the other hand, are comparatively transparent gases, and their radiating properties are small. Moreover, what evidence there is goes to show that the rate of emission of radiation from any gas varies with the pressure. CO_2, for instance, at 400 lb. per sq. in. (abs.) would not have the same radiating effect as the same gas at atmospheric pressure, even if the temperature did not alter. Tyndall,[1] working in the sub-atmospheric region, found that the absorption of CO_2 varied from 12 per cent. to 50 per cent. as the pressure was increased from 1 to 30 inches of mercury. Absorption and emission properties, generally speaking, vary in a similar manner.

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[1] "Heat," by J. H. Poynting and J. J. Thomson (1908), p. 229.

[2] Quoted in D.A.P. (1924), vol. i. p. 663, footnote.

[3] See Kaye and Laby, "Physical and Chemical Constants" (1911), p. 65. The D.A.P., vol. i. p. 663, gives $\sigma=1{\cdot}36\times10^{-12}$ instead of $1{\cdot}266\times10^{-12}$ cal. per sec. This may represent more recently determined values, or it may be only a misprint.

[4] "Heat Transmission," Proc. Inst. Mech. E. (1909), p. 924.

The problem is further complicated by the fact that all the CO_2 and H_2O particles in a hot flame may be radiating. Some of this radiation from the surface of the flame will reach the cylinder walls, but some of it from inside the flame will be reabsorbed by other particles of CO_2 and H_2O, whilst a further supply from an inner source may pass out through comparatively transparent air particles.

W. T. David, who is the pioneer in obtaining actual measurements of the degree of radiation from coal gas when exploded in a closed vessel, has published results of his experiments in the form of curves.[2] In these experiments, turbulence was absent, and the time, after the pressure begins to rise, reaches half a second before the temperature drops to 1300° C. (abs.). Three mixtures of coal gas and air are given, the proportions being 15 per cent., 12·4 per cent., and 9·7 per cent. respectively. The following table is deduced from Figs. 2, 3, and 4 in this paper, the average values being computed from the portion of the curves in the first half-second which rise above 1300° C. (abs.). These curves show that throughout this period radiation was present.

Mixture.	15 per cent.	12·4 per cent.	9·7 per cent.
Max. temp. reached in °C. (abs.) .	2,440·0	2,050·0	1,660·0
Time after pressure begins to rise .	0·05	0·08	0·18
Average temp. during period over 1300° C. (abs.)	1,760·0	1,668·0	1,500·0
Max. radiation in calories per sec.	0·6	0·3	0·2
Average rate of radiation during same period in calories per sq. cm. per sec.	2·07	1·50	1·04
Corresponding rate for full black body	12·18	9·83	6·43
Percentage ratio of last two lines .	17·0	15·25	16·2

The last line but one is calculated from the Stefan-Boltzmann law :

$$H=1{\cdot}27\times10^{-12}(T_1^4-T_2^4)$$

where T_2^4 is considered to be negligible, and T_1 is the average height of David's temperature curves over 1300° C.

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[1] "Heat a Mode of Motion," 6th ed. (1880), p. 415.

[2] "Heat Loss in Gas Engines," by W. T. David. Proc. Inst. Mech. E., October 1924, p. 745.

The composition of the products of combustion of these three mixtures was as follows:

	15 per cent. mixture.	12·4 per cent. mixture.	9·7 per cent. mixture.
N_2 per cent.	71·5	76·6	81·6
CO_2 ,,	8·5	7·0	5·5
H_2O ,,	20·0	16·4	12·9

Since artificial turbulence was absent in these experiments, and the time during which the temperature remained over 1300° C. (abs.) was longer than would occur in any single explosion stroke of an internal-combustion engine (except, perhaps, very slow-running marine oil engines) these values for radiation, which David says in his paper may also be taken to vary as the fourth power of the absolute temperature, may be looked upon as representing the lower limit of radiation. Hence

$$\text{upper limit} = 1{\cdot}27 \times 10^{-12}(T_1^4 - T_2^4)$$
$$\text{lower limit} = 0{\cdot}204 \times 10^{-12}(T_1^4 - T_2^4)$$

where an average ratio of 16 per cent. of total radiation is assumed from David's results, and the units are calories per sq. cm. per sec. T being in °C. abs.

The various reports of the Gaseous Explosions Committee of the British Association[1] contain much information on the effects of radiation in explosion vessels and combustion chambers, but the two main factors which can be stated in the light of our present knowledge follow from what has already been considered here.

(*a*) The increase in radiation as approximately the fourth power of the absolute temperature, means that richer (*i.e.* hotter) mixtures radiate much more, in proportion, than weak mixtures. This sets a definite limit to the maximum mean effective pressure which can, at present, be conveniently utilised in an internal-combustion engine.

(*b*) The radiation from a large flame being greater, in proportion, than that from a small flame, means that there is a limit to the size of the combustion chamber and therefore of the cylinder of internal-combustion engines.

The first restriction results in an upper limit of brake mean effective pressures of the order of 100 lb. per sq. in., except in racing-car petrol engines of relatively small cylinder volume.

The second restriction means that 2000 H.P. is about the economic limit of power to be developed in one cylinder at the present time.

Temperature Stresses in Metals.—So far the effects of temperature in an internal-combustion engine have been considered chiefly from the point of view of thermal efficiency, but temperatures may be reached which will have a definite effect on the materials used in construction. All metals expand with increasing temperature, and if free play is not given to such expansion, stresses will be set up. But the nature of metals varies with high temperatures, and this introduces a further complication.

The Coefficients of Linear Expansion.—The linear expansion of metals over the range of temperatures found in an internal-combustion engine may be taken as directly proportional to the temperature. Hence a bar of length l at $t_1°$, becomes

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[1] Notably the 2nd (1909), the 3rd (1910), and the 5th (1912) Reports.

$$l\{1+\alpha(t_2-t_1)\}$$

at $t_2°$, where α is the coefficient of linear expansion. For practical purposes, the following values of α per °C. may be used:—

Mild steel	$10{\cdot}0\times10^{-6}$
Cast iron	$10{\cdot}0\times10^{-6}$
Aluminium	$25{\cdot}5\times10^{-6}$
Brass	$18{\cdot}5\times10^{-6}$

For special steels used in the construction of valves a modern book of tables, such as Kaye and Laby, should be consulted.

Stress set up due to Change of Temperature.—If the metal were rigidly fixed, so that it could not expand or contract with changes of temperature, then the material would be stressed by the strain due to $\alpha(t_2-t_1)$.

This stress may be calculated per unit area of cross section from the relation

$$E\alpha(t_2-t_1)$$

where E=Young's modulus of elasticity.[1]

The late Professor B. Hopkinson[2] discussed the case of a piston crown, which had temperatures under normal working conditions of 340° C. at the centre, and 170° C. at the periphery above the mean jacket temperature. He shows by calculation that there will be a contraction in diameter of such a crown, considered as a flat disc, which is half that corresponding with the fall in temperature from the centre (assumed constant) to the rim.

If the whole piston face were heated uniformly through 340° C. it would expand

$$340\times10{\cdot}0\times10^{-6}=0{\cdot}0034$$

of its diameter (which was $11\frac{1}{2}$ inches in this case).

The contraction caused by the cooling of the rim to 170° C. below the temperature of the centre would be superimposed on this expansion. Since this contraction may be taken as half that due to 170° C. fall in temperature, it will amount to $\frac{170}{2}\times10{\cdot}0\times10^{-6}=0{\cdot}00085$ of its diameter. Hence the net expansion will be

$$0{\cdot}0034-0{\cdot}00085=0{\cdot}00255 \text{ of the diameter.}$$

There will be a tensile stress set up in the rim corresponding to this extension. Taking E for cast iron as 6,700 tons per sq. in., the tensile stress in this case would amount to $6700\times10{\cdot}0\times10^{-6}\times\frac{170}{2}=5{\cdot}7$ tons per sq. in.

Hopkinson further shows that with his heaviest charges the temperature at the centre rises to 440° C., and at the edge to 220°

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[1] See "Strength of Materials," by A. Morley (1917), p. 59.

[2] "On Heat Flow and Temperature Distribution in the Gas Engine," by Bertram Hopkinson, Proc. Inst. C.E., vol. clxxvi. (1908–9), p. 224.

above the mean jacket temperature. In that case the tension in the rim is 7½ tons per sq. in. If it were assumed that the diameter of the piston were doubled, Hopkinson shows that the temperature at the centre would reach 600° C., and a rim stress of the order of 10 tons per sq. in. would result. This is on the assumption that E for cast iron does not alter at such a temperature. He points out, however, that this calculation of stress applies to a plain disc of uniform small thickness. Actual pistons are not always plane over the whole area, and are frequently reinforced by thickening the metal at the rim as well as by ribs inside. These features tend to reduce the stress by distributing it over a larger area.

Reduction in Tensile Strength of Metals at High Temperature.—There is another factor in this connection which has not been considered as a rule, and that is the effect of high temperatures on the physical properties of metals. Professor F. C. Lea and O. H. Crowther have shown[1] that the modulus of elasticity of mild steel, which was $29{\cdot}5 \times 10^{-6}$ lb. per sq. in. at normal temperature, decreased successively to $28{\cdot}0 \times 10^{6}$ at 275° C. to 26×10^{6} at 375° C., to $23{\cdot}0 \times 10^{6}$ at 450° C., and to $12{\cdot}5 \times 10^{6}$ at 600° C. The ultimate breaking strength, on the other hand, first rises with temperature from 28·5 tons per sq. in. at normal temperature to 39 tons at 250° C., and then falls away to 30 tons at 400° C., dropping to only 9 tons at 600° C. Moreover, the metal tends to get more brittle as the temperature is increased.

Cast iron, which is not a homogeneous material like steel, is much more uncertain in its behaviour at high temperatures. There is, at present, no information on this point beyond the fact that it appears to retain its strength (without, however, getting stronger like mild steel) up to about 400° C., but falls away to about half strength at 600° C. If the modulus varies in a similar manner, the temperature stresses per unit thickness, as calculated by Hopkinson, would be correspondingly reduced.

Growth of Cast Iron at High Temperatures.—When cast iron is heated, and particularly when it is subjected to repeated heatings, it may increase in bulk to a considerable extent. The experience of the steam-turbine designer shows that it is unwise to subject cast iron to a steady temperature of much more than 280° C.

Again, Rugan and H. C. Carpenter[2] found that samples of commercial grey iron (3·14 per cent. C, 0·96 per cent. Si) when heated ninety-nine times for four hours, at temperatures varying from 850° C. to 950° C., increased in volume to the extent of 36·80 per cent. In other samples they found that growth commenced at about 650° C., and they considered that the increase found by the steam-turbine designer at about 300° C. was due to a different phenomenon. The metallurgist, quite rightly, makes a distinction between true growth, and an increase in size due to causes not inherent in the constitution of the iron; but to the engineer all forms of enlargement are harmful, and the term "growth" will be used here in its looser sense. It appears that growth may be due to the following causes:—

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[1] "The Change of the Modulus of Elasticity and of other Properties of Metals with Temperature," by F. C. Lea and O. H. Crowther, British Association Report (1914), reprinted in *Engineering*, vol. xcviii. (1914), p. 487.

[2] "The 'Growth' of Cast Irons after Repeated Heatings," by H. F. Rugan and H. C. H. Carpenter, Journal of Iron and Steel Institute, vol. 80 (1909), p. 42.

(*a*) Expansion of minute particles of dissolved gases held in the iron.

(*b*) The conversion of combined to free carbon.

(*c*) The action of the silicon and phosphorus content of the iron in a way which is not clearly understood.

(*d*) In the case of the internal-combustion engine, the penetration of gas and oil under pressure by way of the graphite plates.

The difficulty in diagnosing any particular case of failure, which may be due to growth, is so considerable that it is wiser to refer the matter to a trained metallurgist. Information on the subject may be found in several chapters of Sir W. H. Hatfield's "Cast Iron in the Light of Recent Research" (1918). Hurst[1] has examined cracked piston crowns from Diesel engines, and found liquation of the phosphide eutectic, which meant that the temperature must have been in the region of 900° C. This is considerably higher than the 600° C. which Hopkinson predicted as being the probable maximum permissible temperature of the centre of a piston crown (see p. 191). In the former case, however, radial cracks were found at the centre, whereas in Hopkinson's calculation the piston would fail at the rim.

Although there appear to be various causes of so-called growth in cast iron, it has been found that the addition of steel to cast iron entirely eliminates this phenomenon. Various suggestions have been made from time to time as to a suitable composition of such semi-steel castings. The following table shows three of these proposals:

Authority	Carpenter.[2]	Cook.[3]	Rhead.[4]
Total carbon . . .	2·6	3 to 3·2	3·25 per cent.
Manganese	1·6	under 0·5	at least 0·7 ,,
Silicon	0·6	1 to 1·2	up to 1·2 ,,
Phosphorus	—	under 1	0·6 ,,
Sulphur	—	1·12	0·1 ,,

The presence of manganese appears to hinder the separation of the carbon and to toughen the alloy. Cook, however, points out that it tends to the formation of brittle spikey crystals which break off under rubbing, and therefore it is not suitable for castings which are required to have good wearing properties when working under heated conditions. Carpenter's mixture is stated to show no growth after 150 heatings to high temperatures. It is, however, recommended for high-pressure steam valves and turbine casings.

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[1] "Cast Iron with Special Reference to Engine Cylinders," by J. E. Hurst, Manchester Association of Engineers, December 9, 1916.

[2] Quoted in *Power*, vol. 60 (1924), p. 668.

[3] "Diesel Engine Castings," by F. J. Cook, Trans. N.E. Coast of Eng. and Shipbuilders, vol. 36 (1910), p. 175.

[4] Quoted in "Modern Practice in Heat Engines," by Telford Petrie (1922), p. 244.

The percentage addition of steel scrap depends, to some extent, on the analysis of the pig and iron scrap with which it is melted. Mr. E. L. Rhead's mixture is based on the following proportions: selected shop scrap, 50 per cent.; a good Mn iron pig, 30 per cent.; mild-steel scrap (such as boiler plate not less than $\frac{3}{8}$ in. thick), 20 per cent. Mr. Cook considers that more than 15 per cent. of mild steel will tend to make the metal too hard. The main object of this addition of steel is to lower the silicon and phosphorus contents and to strengthen the casting. These semi-steel castings are more difficult to obtain in the foundry than pure cast iron ones, and considerable care is necessary to avoid internal cooling strains.

Water Injection.—When compression ratios were first increased, a method of avoiding pre-ignitions was introduced to keep down the resulting temperature. This consisted in injecting a spray of water into the cylinder, either during the suction stroke or at the commencement of compression. Water has a very high latent heat of evaporation, so that some of the heat which would otherwise go to raise the temperature of the gaseous mixture during compression, is required to turn the water into steam, with the consequence that the gas can actually be compressed to a greater amount before the danger point is reached.

This means, however, that the engine would have to be specially designed with a higher compression than usual, in order to get a better thermal efficiency from this cause. The cooling effect of the water should not, on this hypothesis, improve the economy of the engine if the compression is not altered, but with higher compression, any failure of the water-injection, even for a stroke or two, would lead to serious pre-ignition. Moreover, if the fuel contained any sulphur, as is often the case with town gas or crude oils, the nascent hydrogen and oxygen set free from the water molecule at the high temperature reached after explosion, would combine to form sulphurous acid, and this, if allowed to condense, is very destructive, and would eat away the cylinders and pistons in a very short time.

If due precautions are taken, as for instance in 1904, when Professor Burstall tested a 50 H.P. gas engine with spray injection on the suction stroke, much higher efficiencies are possible than would otherwise be the case.[1] In this test, the engine had a compression ratio of 8·7, and compression pressure exceeded 200 lb. per sq. in. The water entered in a fine spray as part of the air supply. It was found that very little water was required, the total amount being only 0·16 lb. per B.H.P. per hour, and the brake thermal efficiency was as high as 30·8 per cent. The indicated thermal efficiency was 37 per cent. At the date of this test, these were the highest figures ever reached in any kind of heat engine. This was due entirely to the raising of the compression beyond what was thought possible in a gaseous mixture. The makers of this engine published the following statement at the time: "Our system was simply to employ high compression in conjunction with the injection of a small water spray. We claim no special economy for the use of water. We find that as we use it, it prevents pre-ignition and any undue rise of temperature in the cylinder, making it possible to use gas with a higher percentage of hydrogen at a very high compression. It also improves general conditions by reducing temperatures throughout the cycle of operations."

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[1] Some details of this test may be found in "The Internal-Combustion Engine," by H. E. Wimperis, 4th ed. (1922), p. 128.

Water injection for gas engines, however, was subsequently discontinued, partly on account of the difficulties and dangers outlined above, and partly because of its possible misuse by customers. It is, however, still used in a number of oil engines to prevent "knocking" at full loads, *i.e.* when running on rich mixtures.[1]

Some interesting experiments were carried out in 1920 at the Manchester College of Technology by W. J. Walker and R. Koivulehto,[2] on an experimental four-stroke gas engine, which had a cylinder $8\frac{1}{2}$ in. in diameter by 20 in. stroke. This engine developed an average brake horse power of 18 at 220 r.p.m.

Their object was to ascertain how the brake thermal efficiency and the heat losses varied with:

(*a*) The quantity of water injected.
(*b*) The compression ratio.
(*c*) The richness of the mixture.

Tests were also made to determine the best period of admission for the water injection; but it was found that the actual portion of the cycle during which injection took place had little effect on the results.

To test this, water was injected during the following periods of the cycle:

(*a*) From 0·45 to 0·98 of the compression stroke.
(*b*) From 0·02 to 0·40 of the compression stroke.
(*c*) From 0·20 to 0·70 of the expansion stroke.
(*d*) From 0·80 of the compression to 0·30 of the expansion stroke.

Each test consisted of a full-load brake trial of 20 minutes' duration, and in all some 100 trials were run.

The method of water injection was somewhat similar to the oil spray of a solid-injection engine. By means of a small hydraulic accumulator working off the town mains, water at a pressure of 550 lb. per sq. in. was admitted into the centre of the back end of the cylinder through a single-holed nozzle. The quantity injected was varied by using different-sized nozzles of $\frac{1}{64}$, $\frac{3}{128}$, and $\frac{1}{32}$ in. diameter, respectively. The period of admission was controlled by an adjustable cam working off the half-motion shaft.

The compression ratio of the engine was altered by fitting plates of different thicknesses to the piston face. In this way compression ratios of 5·2, 5·7, 6·1, and 8·92 were obtained. Only a few trials were run with a compression ratio as high as 8·92. It was not found possible to run any rich-mixture tests at this compression without water injection, owing to the heavy knock which developed, and after some seven trials had been run with the period of injection as (*a*), this compression ratio was abandoned.

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[1] For an example see the D.A.P., vol. i. (1922), p. 332.

[2] A full account of these tests will be found in a thesis by R. Koivulehto, which may be consulted in the college library. See also "Experiments on Water Injection in a Gas Engine, and its Effects upon Heat Distribution," by W. J. Walker and R. Koivulehto, Proc. Inst. Civ. Eng., vol. ccx. (1919–20), Pt. II., p. 394.

Two different mixtures were used in each set of experiments, a rich mixture of approximately 6 to 1, and a normal mixture of 7 to 1. Town gas was used with the following average analysis CO_2 4·1 per cent., O_2 0·1 per cent., C_2H_4 2·0 per cent., CO 19·5 per cent., CH_4 17·5 per cent., H_2 50·3 per cent., and N_2 6·5 per cent. The calorific value of this gas was measured for each trial; it had an average lower value of 415 B.Th.U. per cu. ft.

Fig. 58 shows the relation obtained between the brake thermal efficiency and the quantity of water injection over the period between 0·45 to 0·98 of the compression stroke. Very similar curves were obtained over the other periods mentioned above. In no case was the thermal efficiency improved when water was used, in fact there was always a gradual decrease in its value as shown in these curves

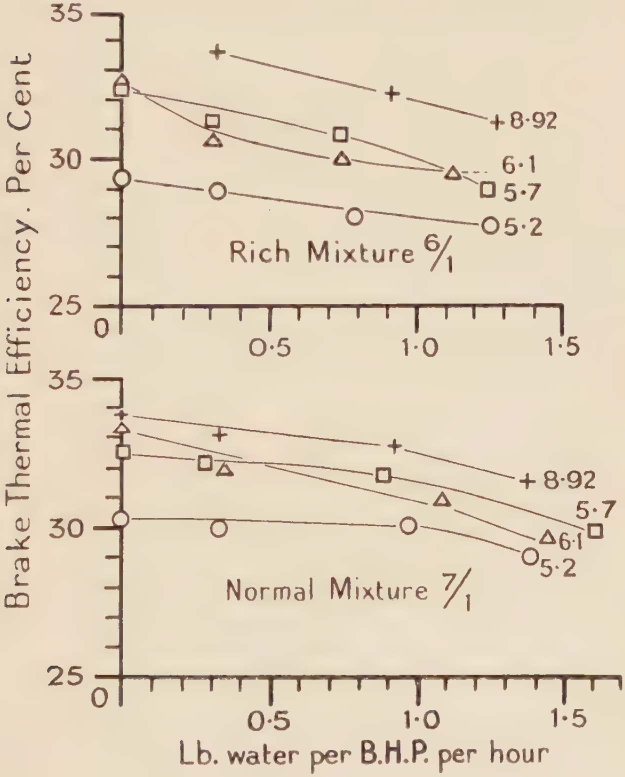

FIG. 58.

Fig. 59 shows the relation between the heat diverted to the jacket and that shown in the exhaust for the same period of admission. Here, again, similar results were obtained for the other periods of water admission. The effect of introducing water shows very clearly how jacket loss may be turned into exhaust loss at a slight sacrifice of thermal efficiency. This is the most important point brought out by

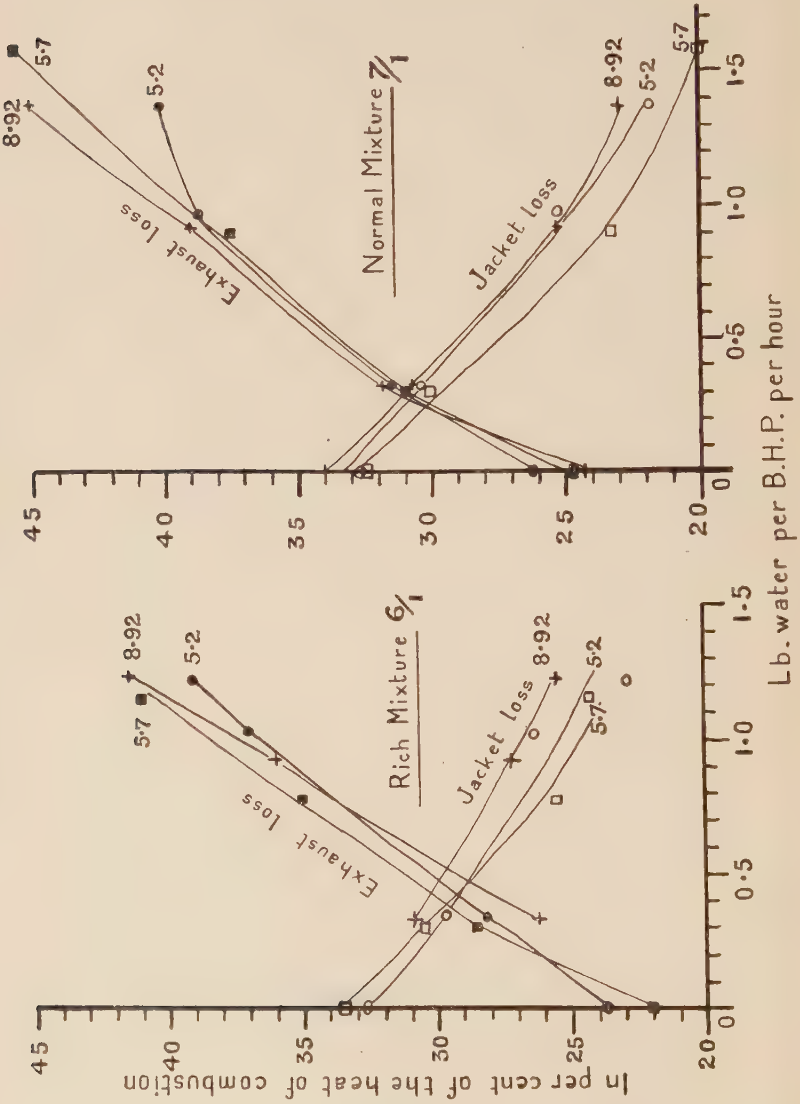

Fig. 59.

these trials, and shows that if any attempts are to be made to use the heat in the exhaust either in waste-heat boilers or in some form of exhaust-gas turbine, considerable control may be obtained by the use of water-injection in the prime-mover.

In these tests, the engine was in no way damaged, in fact there was a distinct cleaning effect of water injection, and a marked absence of any carbon deposit. This same effect has been noted by most of the other experimenters on this problem.[1] Care was always taken, however, to run the engine dry for ten minutes or so after using water injection to make sure that no water was left in the cylinder whilst the engine was standing cold.

Hopkinson[2] carried out another form of water injection to which frequent reference is made. The essential feature of his method was to introduce water in the form of coarse drops (not as a fine spray) during the period when the piston was near its in-stroke, and therefore when the combustion chamber was small. Hopkinson found that he could dispense with a water-cooling jacket altogether. He also obtained about 5 per cent. better consumptions when using rich mixtures with water injection in place of water jackets, whereas with weak mixtures the reverse proved to be the case. The quantity of water used was about 2·4 lb. per B.H.P. per hour, which is much more than would be advisable for spray injection on the suction or compression stroke.

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[1] See, for instance, the American National Advisory Committee for Aeronautics Report No. 45, of 1918, where experiments on petrol engines showed that water injection between 0·1 and 0·5 lb. per B.H.P. per hour produced a softening and reduction of carbon deposit up to about 25 per cent.

[2] "A New Method of Cooling Gas Engines," by B. Hopkinson, Proc. Inst. Mech. E. (July, 1913).

CHAPTER XII

THE POSSIBILITIES OF DEVELOPMENT

Historical.—To attempt to forecast the future of any line of development is as fascinating as it is dangerous. The internal-combustion engine, starting in the sixties of the last century, became a sound practical proposition under Otto (1876) and Clerk (1880).

The two principles of the Otto four-stroke cycle and of the Clerk two-stroke cycle remain to this day the foundations upon which the designs of most internal-combustion engines rest.

Most of the successful inventors of this period built upon these foundations, adding a little here and a little there, and obtaining their information chiefly by the time-honoured method of full-scale experiments. The Priestmann brothers succeeded in making the gas engine of their day work with oil vapour. H. Akroyd-Stuart went one better and formed his oil vapour in a chamber connected with the working space of the engine. In this way the germ of the idea of the hot bulb was introduced. But the Hornsby-Akroyd engine still functioned as a gas engine, in that oil vapour and air, a combustible mixture, were drawn in on the suction stroke and compressed together. Step by step numerous inventors introduced the oil later and later during the cycle, until the true hot-bulb engine was evolved, which compressed air only, and injected its oil fuel just before the explosion stroke.

Before this happened, however, another genius, Rudolph Diesel, had started to reason from first principles, and evolved a new type of engine which is always known by his name. Diesel was so original in all that he did that he attempted, in the first place, to evolve an engine to work with solid fuel, such as coal dust, on the Carnot cycle. It was only after an engine had actually been constructed that the idea was abandoned.

The Carnot cycle requires such enormous pressures to obtain anything like a reasonable M.E.P., that its use for internal-combustion work is still probably impracticable, but it might well be worth some one's while to revive the idea of using powdered fuel in an internal-combustion engine now that so much more is known about its use under boilers. The introduction of coal dust in small enough quantities would require considerable mechanical ingenuity, but the use of blast air, as in a Diesel, and the provision of a hot bulb to act as a reverberatory chamber and ensure complete combustion, would seem to be a possible arrangement. The advantages of such an engine, particularly for land work, would be considerable, since the price of the fuel would only be about one-quarter that of crude oil.

The hot-bulb engine, as it is known to-day, reached its final form after Diesel had exploited his engine; in fact, some makers still refer to it as a semi-Diesel. But the hot-bulb engine works on a more efficient cycle than the Diesel, a fact which is not usually apparent because the compression ratios are different in the two cases. For a given compression ratio, the fact remains that the Otto (or Clerk) cycle gives a higher mean effective pressure than the Diesel cycle.

Up to the present, all gas and petrol engines are limited in their compression by the possibility of pre-igniting a combustible mixture. And yet Gibson and Baker[1] obtained a brake mean effective pressure of the order of 120 lb. per sq. in. in an air-cooled cylinder, 100 mm. bore, with a compression ratio of 5·5 when running at 1800 r.p.m. A Diesel engine rarely exceeds 80 lb. per sq. in. brake mean effective pressure. Of course temperature effects, and particularly the relative increase of radiation with the size of the flame, play an important part in limiting such pressures, and a careful balance between all the limiting factors will have to be struck when considering any possible development. The fact that Gibson's cylinder was air cooled, however, would seem to indicate that a margin still exists in this direction, since air cooling is not as efficient as water cooling.

Some day, perhaps, successful attempts will be made to introduce petrol or gas into the cycle at, or near, the end of the compression stroke, thereby enabling higher compression ratios, and therefore higher brake thermal efficiencies to be obtained than in any existing petrol or gas engine.

Mechanically, this problem would not be difficult, since in the case of petrol some form of spring-operated pump could be evolved on the lines of a solid-injection engine sprayer. In the case of gas, a compressor would be needed, but a rich gas like town gas would compress to any required extent without danger of explosion, provided air was not present in sufficient quantity to form an explosive mixture.

The water-injection tests of the Manchester College of Technology perhaps point to a way in which such a compressor could be driven. By diverting the bulk of the heat loss to the exhaust, an exhaust-gas turbine on the Rateau principle,[2] or a small waste-heat boiler generating steam to drive a compressor might prove to be within the margin for economy, since this regenerative principle has already achieved marked success in the case of the Still engine.

Alternatively, the Diesel principle of driving the compressor off the crankshaft might be tried. Again, the waste heat might be utilised to drive the Diesel-engine compressor, and so improve the mechanical efficiency of this type of engine.

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[1] "Exhaust-Valve and Cylinder-Head Temperatures in High-Speed Petrol Engines," by A. H. Gibson and H. Wright Baker, Proc. Inst. Mech. E., December (1923), Table I., p. 1056.

[2] For details of this turbine as applied to aeroplane engines, see "The Use of the Turbo-Compressor for attaining the Greatest Speeds in Aviation," by A. Rateau, Proc. Inst. Mech. Eng. (June, 1922), p. 795.

With the exception of the Still engine, the possibilities of extracting further useful work from the exhaust and jacket losses cannot, as yet, be said to have been fully investigated. Except in special localities, such as the North-East Coast of England, the present feeling is that such developments would not pay; but this feeling was just as prevalent before the Still engine was produced. Consider the simple case of an internal-combustion engine developing 100 H.P., with 30 per cent. efficiency. The curves on p. 196 show that 30 per cent. of the total heat may easily be diverted to the exhaust. This represents $100 \times 2546 = 254{,}600$ B.Th.U. per hour.

It takes about 1,160 B.Th.U. to raise 1 lb. of steam at 100 lb. per sq. in. pressure from cold-feed water. If this exhaust heat were passed into a waste-heat boiler with an efficiency of conversion of 50 per cent., then the number of lb. of steam which could be generated per hour at 100 lb. pressure

$$= \frac{254600}{1160} \times 0{\cdot}5 = 110 \text{ lb.}$$

If this steam were utilised in a small non-condensing steam engine using 20 lb. of steam per B.H.P. per hour., this heat recovery represents $5\frac{1}{2}$ B.H.P. per 100 B.H.P. on the main engine, or $5\frac{1}{2}$ per cent. It is, however, possible to get waste-heat boilers with a higher efficiency than 50 per cent. Moreover, the jacket water could be used to warm the boiler feed to, say, 100° F., and the exhaust losses by means of water injection could be increased to 40 per cent. In this case the following holds:

$$\frac{254600}{1120} \times \frac{4}{3} \times \frac{0{\cdot}6}{20} = 9 \text{ B.H.P., or 9 per cent.}$$

That such figures are not beside the mark is shown by some tests on the Still engine,[1] where 12 per cent. of the brake horse power of the engine is recovered on the steam side. In these tests, an engine of 329 nett B.H.P. produced 807 lb. of steam at 100 lb. per sq. in. pressure, which generated about 40 B.H.P. in the engine.

If only 10 per cent. of the power of a Diesel were recovered in this way, the mechanical efficiency could be raised from 75 per cent. to 82 per cent. This can be seen from the following simple calculation: 100 B.H.P. at 75 per cent. efficiency represent 133 I.H.P. Add 10 B.H.P. and the efficiency becomes $\frac{100+10}{133} = 82$ per cent. In a similar way 78 per cent. efficiency becomes 86 per cent., which corresponds to that of the solid-injection engine where no air compressor is needed.

This method of generating steam, however, is not of much use for process work or heating the factory, on account of the small quantities produced. A 500 H.P. gas engine installation might generate 1000 lb. of steam per hour; but an extraction steam engine[2] could easily pass five times as much and still prove economical.

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[1] Quoted in *Engineering*, vol. cxii. (1921), p. 344.

[2] See, for instance, *Engineering*, vol. cxiv. (1922), p. 584.

Further Progress—Marine Development.—To return to the historical development of the internal-combustion engine. The Diesel engine became so prominent that it was almost placed in a different category to the ordinary internal-combustion engine, and developed along its own lines. Many difficulties arose, chiefly connected with heating stresses, and these were successively overcome by a series of brilliant designers. Powers increased until some 2000 H.P. was generated in a single cylinder. But quietly, in the background, another type of oil engine was being evolved. Like a good many other inventions, the Great War gave the necessary impetus, and now the solid-injection engine more than holds its own in economy with the Diesel engine.

As has been seen, there are many difficulties connected with its development, and the type has probably not yet reached finality, but its inherent simplicity, together with its great economy, should eventually make the solid-injection oil engine outstrip all rivals that can at present be seen. W. J. Still, the most original thinker in the internal-combustion world since Diesel, uses solid injection in his engine because, as he says, " fuel-injection troubles may prevent reliability, and I will say at once that our use of a ' solid-injection ' system was inspired by this consideration." [1]

The use of oil engines at sea—some two million horse power is already afloat—has greatly added to the interest in the development of internal-combustion engines. New problems are introduced, and their solution, at the present time, cannot be said to have been standardised. Only full-sized trials on a large scale can prove the outstanding merits (if any) of one particular system. In this connection, the reports [2] of the Marine Oil-Engine Trials Committee of the Institution of Mechanical Engineers will be followed with great interest. Amongst other outstanding novelties, serious attempts are being made, at the present time, to introduce the double-acting principle into crude-oil engines, with the idea of saving weight for power as much as possible. Others are using the opposed piston arrangement, with or without the Fullagar balancing effect. The solid-injection principle first came into prominence in connection with the driving of submarines, and now it is being applied to other types of marine oil engines. The whole problem of the application of the oil engine to marine propulsion is in a most flourishing state, and future developments in this connection are bound to be interesting.

New Cycles.—From time to time internal-combustion engines appear which work on different cycles. The Otto, or constant-volume, cycle is still the most common, but Diesel produced another type of cycle, whilst, as far back as 1886, Atkinson succeeded in extending the " toe " of the Otto cycle. Humphrey, in his famous water pump, carried the expansion right down to atmospheric pressure. Then oil engines of the solid-injection type appeared, which worked on the dual-combustion cycle, and Walker went even further and patented a cycle with dual combustion and dual exhaust.[1] This cycle is the same as that on which the uniflow steam engine functions, but so far it has not been applied definitely to internal-combustion engines.

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[1] " The Type of Still Engine required for Marine Service, with Special Reference to the Scott-Still Engine," by W. J. Still, Trans. N.E. Coast Inst. of Eng. and Shipbuilders, vol. 40 (1924), p. 408.

[2] The first report was read in London, November 21, 1924.

All these cycles exhaust at about atmospheric pressure, and the only attempt to extract work in the sub-atmospheric region was that of the Otto and Langen free-piston engine of 1866. The theoretical value of the efficiency of this cycle, considered as half a Carnot cycle, could be very high, but the relative efficiency or coefficient of performance was miserable. This brings out a very important point in the consideration of heat cycles. The cycle itself may show a very high efficiency, but the important factor is not so much this unattainable ideal as the degree of approach which the actual engine makes to it. For a given compression ratio, the Otto cycle is more efficient than the Diesel, but in actual performance the Diesel engine more nearly approaches its ideal, and therefore shows a greater relative economy in fuel consumption.

The problem of one new cycle, with the above reservation, appears to come to this: How far can the sub-atmospheric region be exploited so as to add to the mean effective pressure without increasing the temperature effect? The steam engine solves this problem easily, by the simple expedient of creating a vacuum by condensing steam; but even here an air pump is required. The internal-combustion engine cannot condense its products of combustion, so that it would require some form of suction pump to produce a vacuum. Such an arrangement would add very considerably to the complications of the heat engine, and it remains to be proved whether the extra amount of work recovered is greatly in excess of the extra energy expended in obtaining a vacuum. There is, however, an alternative method of dealing with the sub-atmospheric region, and that is the one already referred to under waste-heat recovery. This is, as in the Still engine, a transfer of a portion of the exhaust heat into steam which can produce (if required) the necessary vacuum.

Super-compression and Supercharging.—Another method of improving the output of an internal-combustion engine, without, however, introducing a new cycle, is to increase the charge of air or of inert gas before compression begins, so as to lift the adiabatic compression curve, and reach a higher compression pressure without increasing the compression temperature. This is known as *super-compression*, and was first introduced by Sir Dugald Clerk[2] in 1904. Clerk showed that an ordinary 4-stroke cycle gas engine, which had a thermal efficiency of 27·7 per cent., could have its thermal efficiency raised to 34·4 per cent. by the introduction of a charge of air near the end of the suction stroke, which increased the pressure, before compression, to about 22 lb. per sq. in. (abs.). This had the effect of increasing the weight of gas and air present by some 40 per cent. Clerk found that the maximum temperature, in either case, was about

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[1] British Patent 118934 of 1918.

[2] "Internal-Combustion Motors," by Dugald Clerk, 12th "James Forrest" lecture. Proc. Inst. Civ. Eng., vol. clviii. (1904), p. 302.

1200° C., whilst, with the addition of the extra air, the temperature of compression was actually reduced. The mean effective pressure was increased about 20 per cent.

Super-compression is also reported to have been applied to Diesel engines of the Junkers type.[1] In this case it was obtained by partially throttling the exhaust, with the result that the scavenging air is left in the cylinder at a higher pressure; temporary overloads of 50 per cent. have been met in this way.

Supercharging, which is not quite the same thing as super-compression, is coming into use in connection with racing motor-cars. In fact, some continental makers now give three ratings; the first one being the taxable horse power; the second, the maximum horse power without supercharging; and the third, which may be some 30 per cent. higher than the second, when a small Root's blower is connected in by means of a friction clutch to the crankshaft, and a supercharge added.

The whole question of supercharging the Otto cycle has been very ably set out by Georges Funck,[2] who shows the application of Stodola's entropy diagram to this problem. He defines the object of supercharging as follows: "To consume a greater quantity of fuel per cylinder filling than would normally be possible, and from the resulting additional heat units set free during combustion to extract a greater H.P. output." It will be noted that this definition differs from that for super-compression, which aims at increasing the weight of the charge without necessarily increasing its richness.

The Gas Turbine.—The subject-matter of this book has been entirely confined to the reciprocating internal-combustion engine, partly because the author has had no direct experience with the rotary type, and partly because this latter type, commonly known as the gas turbine, cannot yet be said to be a serious rival of the reciprocator. But the gas turbine cannot be omitted from any chapter which attempts to deal with the possibilities of development. Information on the subject is not, as yet, very accessible to British engineers. Probably the most convenient consists in a series of seven articles on "The Gas Turbine in Theory and Practice," which were published in *The Engineer* in 1923.[3] The greatest authority on the subject is A. Stodola, who has a section of eighty-five pages in the 5th edition of his book on "Steam and Gas Turbines."[4] In this, Professor Stodola develops the variable specific heat theory of gases, and provides a complete entropy chart drawn up on the kg.-mol. basis. He also describes various designs and attempts which have already been made to produce gas turbines, and adds a chapter on the rotary compressor.

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[1] *Zeit. d. Verein. deutsch. Ing.*, vol. lxi. (1917), p. 283. There is a summary of this method in "Modern Practice in Heat Engines," by Telford Petrie (1922), p. 238

[2] "The Supercharging of Internal-Combustion Engines," by Georges Funck, *The Automobile Engineer*, vol. 9 (1919), pp. 318, 377, and 421.

[3] *The Engineer*, vol. 135 (1923), pp. 466, 490, 515, 557, 583, 595, and 630.

[4] "Dampf- und Gas-Turbinen," by A. Stodola, 5th ed. (1922), pp. 968–1053.

The theory of the gas turbine is also developed by W. Schüle,[1] who confines himself chiefly, however, to the Holzwarth turbine, which, so far, has made more practical progress than any other type. Holzwarth, himself, has written a book on his turbine,[2] which contains much constructional information, but his analytical deductions and calculations are difficult to follow, and the results of his tests have been published in such a form that it has not been possible to understand their significance. The author has been privileged to see an advance copy of a paper on gas turbines in English, by Max König,[3] which contains a clear summary of Stodola's work, a historical review, and much information on the mechanical and thermal difficulties of their design.

The Cycles of the Gas Turbine.—The actual development of gas turbines has been along two main lines, known as the constant-pressure and the constant-volume-explosion types. In a strict sense, the gas turbine has no cycle, since it works by a series of explosions on a time basis. But, just as it has been found convenient to refer the steam turbine to the Rankine cycle of the reciprocating steam engine, so is it of value to consider what the ideal cycles of a gas turbine may be.

Most authorities postulate the Joule or Brayton cycle for constant-pressure gas turbines and the Humphrey-pump cycle for constant-volume explosion type, but with this difference: that, in both cases, the compression is isothermal and not adiabatic. This is done because the operation is carried out externally in a gas turbine and the ideal cycle for the gas compressor has an isothermal for the compression line.

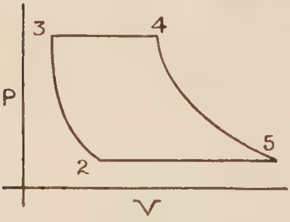

FIG. 60.—Constant-pressure Cycle.

Isothermal compression, as a moment's consideration will show, means that the line 3 to 4, in Figs. 60 and 61 below, will be longer than with adiabatic compression if the point 4 is to occupy the same position on the diagram in both cases. In other words, more heat has to be received in the former case. Alternatively, if the line 3 to 4 is kept the same in both cases, the positive work done will be less for the same amount of heat taken in. It therefore seems that the true Joule cycle for constant-pressure and the Humphrey-pump cycle for constant-volume explosion, both with adiabatic compression, may equally well be used, in the light of present experience, for an analysis of the possibilities of the gas turbine.

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[1] "Technische Thermodynamik," by W. Schüle, Pt. II. (1920), pp. 354–389.

[2] "Die Gas Turbine" (1911), by H. Holzwarth. See also an article by the same author in *Zeitschrift d. Vereines deutscher Ing.*, vol. 64 (1920), p. 197.

[3] "Gas Turbines," by M. Koenig. Trans. North-East Coast Institution of Engineers and Shipbuilders, 24 April, 1925. See *Engineering*, vol. cxix. (1925), p. 747.

The following papers have also appeared since this paragraph was written: "The Internal-Combustion Turbine," by W. J. Goudie, Proc. Institution of Engineers and Shipbuilders in Scotland, 16 Jan. and 24 March, 1925. "The Internal-Combustion Turbine," by T. B. Morley. Trans. Manchester Association of Engineers, 27 Feb., 1925.

The Joule Cycle (Armengaud Turbine).—The constant-pressure type has a cycle, on this basis, like the Brayton engine of 1872; that is to say, it follows the Joule cycle, with constant-pressure exhaust as in Fig. 60.

For any given temperature range T_4 to T_2, this cycle has a ratio of compression $\left(\frac{V_2}{V_3}\right)$, which gives maximum net work, as may be seen from the following analysis.

For unit efficiency the heat received (H)

$$H = C_p(T_4 - T_3)$$

and the heat rejected (h)

$$h = C_p(T_5 - T_2)$$

If the relative efficiency of the turbine is η_1 and the efficiency of the (external) compression η_2, then the heat available for useful work may be written

$$H - h = C_p(\eta_1 T_4 - T_3) - C_p\left(T_5 - \frac{T_2}{\eta_2}\right) \quad . \quad . \quad . \quad (1)$$

The ratio of compression $\frac{V_2}{V_3} = r$

Calling, for the sake of brevity, $r^{\gamma - 1} = y$

Then $T_3 = y\frac{T_2}{\eta_2}$

$$\eta_1 T_4 = \rho T_3 = \rho y \frac{T_2}{\eta_2}$$

but $\rho = \frac{\eta_1 T_4}{y} \cdot \frac{\eta_2}{T_2}$

and $T_5 = \rho \frac{T_2}{\eta_2}$

Substituting these relations in equation (1)

$$\begin{aligned}
H - h &= C_p\left(\rho y \frac{T_2}{\eta_2} - y\frac{T_2}{\eta_2}\right) - C_p\left(\rho\frac{T_2}{\eta_2} - \frac{T_2}{\eta_2}\right)\\
&= C_p\left(\frac{\eta_1 T_4 \eta_2}{y T_2} y \frac{T_2}{\eta_2} - y\frac{T_2}{\eta_2}\right) - C_p\left(\frac{\eta_1 T_4 \eta_2 T_2}{y T_2 \eta_2} - \frac{T_2}{\eta_2}\right)\\
&= \frac{T_2}{\eta_2} C_p\left(\frac{T_4}{T_2}\eta_1\eta_2 - y\right) - \frac{T_2}{\eta_2} C_p\left(\frac{T_4}{T_2} \cdot \frac{\eta_1\eta_2}{y} - 1\right)\\
&= \frac{T_2}{\eta_2} C_p\left\{\frac{T_4}{T_2}\eta_1\eta_2\left(1 - \frac{1}{y}\right) + (1 - y)\right\}\\
&= \frac{T_2}{\eta_2} C_p(y - 1)\left(\frac{T_4}{T_2}\eta_1\eta_2\frac{1}{y} - 1\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)\\
&= \frac{T_2}{\eta_2} C_p\left(\frac{T_4 \eta_1 \eta_2}{T_2} - y - \frac{T_4}{T_2} \cdot \frac{\eta_1\eta_2}{y} + 1\right) \quad . \quad . \quad . \quad . \quad . \quad (2a)
\end{aligned}$$

Differentiating equation (2*a*) with respect to y and equating to zero

$$\frac{d(\mathrm{H}-h)}{dy}=\frac{\mathrm{T}_2}{\eta_2}\mathrm{C}_p\left(-1+\frac{\mathrm{T}_4}{\mathrm{T}_2}\cdot\frac{\eta_1\eta_2}{y^2}\right)=0$$

whence, for maximum heat available,

$$y^2=\frac{\eta_1\eta_2\mathrm{T}_4}{\mathrm{T}_2}\quad\cdots\cdots\quad(3)$$

Since $y=r^{\gamma-1}$ this means that there is one ratio of expansion which gives maximum work for any chosen conditions of temperature range and of efficiencies.

Although the point does not arise in connection with gas turbines, it is worth while noting that equation (3) is equally applicable to the Otto cycle. For the reciprocating internal-combustion engine, the compression efficiency η_2 is part of the engine efficiency η_1, so that the condition for maximum heat available for work in the Otto cycle may be written

$$y^2=\eta\frac{\mathrm{T}_4}{\mathrm{T}_2}\quad\cdots\cdots\quad(3a)$$

where η is the relative efficiency of the engine.

Again, if the ratio of expansion

$$\frac{p_5}{p_4}\text{ or }\frac{p_3}{p_2}$$

be denoted by X, then

$$r=\mathrm{X}^{\frac{1}{\gamma}}$$

Hence $$y^2=r^{2(\gamma-1)}=\mathrm{X}^{2\left(\frac{\gamma-1}{\gamma}\right)}$$

so that the best ratio of expansion (X) for a chosen set of conditions may be obtained from the relation

$$\mathrm{X}^{2\left(\frac{\gamma-1}{\gamma}\right)}=\eta_1\eta_2\frac{\mathrm{T}_4}{\mathrm{T}_2}\quad\cdots\cdots\quad(4)$$

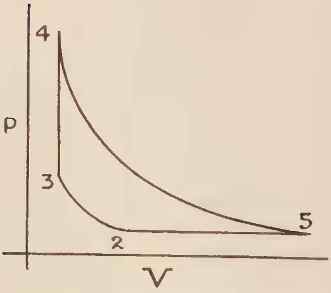

FIG. 61.—Explosion-type Cycle.

The Humphrey Pump Cycle (Holzwarth Turbine).—The constant-volume explosion cycle, Fig. 61, is approximately that on which the Holzwarth turbine functions. In this case, heat is received at constant volume and rejected at constant pressure. Using the notation of the previous paragraph

$$\mathrm{H}-h=\mathrm{C}_v(\eta_1\mathrm{T}_4-\mathrm{T}_3)-\mathrm{C}_p\left(\mathrm{T}_5-\frac{\mathrm{T}_2}{\eta_2}\right)\quad\cdot\quad(5)$$

Unlike the corresponding expression for the constant-pressure cycle (equation 1), the introduction of the efficiencies in this manner cannot be justified on thermodynamical grounds. The following analysis is, therefore, only correct for unit efficiencies, but the discrepancies involved are not as great as would appear at first sight.

$$T_3 = y\frac{T_2}{\eta_2}$$

where $$y = r^{\gamma-1} \text{ and } r = \frac{V_2}{V_3}$$

$$\eta_1 T_4 = \alpha T_3 = \alpha y \frac{T_2}{\eta_2}$$

but $$\alpha = \frac{\eta_1 T_4}{y} \cdot \frac{\eta_2}{T_2}$$

and $$T_5 = \beta\frac{T_2}{\eta_2} = \alpha^{\frac{1}{\gamma}}\frac{T_2}{\eta_2}$$

Substituting these relations in equation (5)

$$H - h = C_v\left(\alpha y\frac{T_2}{\eta_2} - y\frac{T_2}{\eta_2}\right) - C_p\left(\alpha^{\frac{1}{\gamma}}\frac{T_2}{\eta_2} - \frac{T_2}{\eta_2}\right)$$

$$= C_v y\frac{T_2}{\eta_2}(\alpha - 1) - C_p\frac{T_2}{\eta_2}(\alpha^{\frac{1}{\gamma}} - 1)$$

$$= C_v y\frac{T_2}{\eta_2}\left(\frac{\eta_1 T_4 \eta_2}{y T_2} - 1\right) - C_p\frac{T_2}{\eta_2}\left\{\left(\frac{\eta_1 T_4 \eta_2}{y T_2}\right)^{\frac{1}{\gamma}} - 1\right\} \quad . \quad (5a)$$

Differentiating equation (5a) with respect to y and equating to zero

$$\frac{d[H-h]}{dy} = -C_v\frac{T_2}{\eta_2} + \frac{1}{\gamma}C_p\frac{T_2}{\eta_2}\left(\frac{\eta_1\eta_2 T_4}{T_2}\right)^{\frac{1}{\gamma}}\frac{1}{y^{\frac{\gamma+1}{\gamma}}} = 0$$

$$y^{\frac{\gamma+1}{\gamma}} = \frac{1}{\gamma} \cdot \frac{C_p}{C_v} \cdot \frac{\eta_2 T_2}{\eta_2 T_2}\left(\eta_1\eta_2\frac{T_4}{T_2}\right)^{\frac{1}{\gamma}}$$

but $$\frac{C_p}{C_v} = \gamma$$

Therefore $$y = \left(\frac{\eta_1\eta_2 T_4}{T_2}\right)^{\frac{1}{\gamma+1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

This expression should be compared with equation (3) for the constant-pressure cycle, which may be written

$$y = \left(\frac{\eta_1\eta_2 T_4}{T_2}\right)^{\frac{1}{2}}$$

In both cases $$y = r^{\gamma-1}$$

There is no very simple relation between the ratio of expansion $\left(\frac{p_4}{p_2}\right)$ and the compression ratio of the Humphrey pump cycle, but the former can be calculated from the latter when it is remembered that

$$\alpha = \frac{\eta_1 T_4 \eta_2}{y T_2} = \frac{P_4}{P_3}$$

and

$$\frac{P_3}{P_2} = r^\gamma$$

Calling

$$\eta_1 \eta_2 \frac{T_4}{T_2} = A$$

$$\alpha = \frac{A}{r^{\gamma-1}} = \frac{P_4}{P_2 r^\gamma} = \frac{X}{r^\gamma}$$

$$X = \frac{A r^\gamma}{r^{\gamma-1}} = A r$$

where from equation (6)

$$r = (A)^{\frac{1}{(\gamma+1)(\gamma-1)}}$$

Application to Turbine Design.—The expansion takes place through a single turbine wheel in the Holzwarth turbine, which, so far, has proved to be the most successful machine in practice. The work done (U) in expanding from p_4 to p_2

$$U = \int p\,dV = \frac{c^2}{2g} \text{ per lb.}$$

where c is the velocity of the gas through the turbine wheel.

This may be written[1]

$$\frac{c^2}{2g} = \frac{RT_4}{\gamma-1}\left\{1 - \left(\frac{1}{X}\right)^{\frac{\gamma-1}{\gamma}}\right\} \quad . \quad . \quad . \quad . \quad . \quad (7)$$

In the Holzwarth machine, T_4 is about 1500° C. (abs.) and T_2 is atmospheric temperature, say 290° C. (abs.). Assuming for a moment a turbine-wheel efficiency (η_1) of 65 per cent. and a compressor efficiency (η_2) of 60 per cent., and taking $\gamma = 1{\cdot}4$ and R as 96,

then

$$A = 0{\cdot}65 \times 0{\cdot}60 \times \tfrac{1500}{290} = 2{\cdot}015$$

and

$$r = 2{\cdot}015^{\frac{1}{(1{\cdot}4+1)(1{\cdot}4-1)}} = 2{\cdot}076$$

so that the pressure ratios in this case should be

$$X = Ar = 4{\cdot}178$$

The last term in equation (7) then becomes 0·664, and the velocity c is 2,780 ft. per sec.

Now an important factor in the design of turbines is the ratio of the speed of the blades to the speed of the fluid passing through them.[2] This velocity ratio, denoted by the Greek letter α, determines which type of blading is most efficient. The blade speed of any turbine should not exceed 600 ft. per sec., unless special precautions are taken, on account of the mechanical stresses set up. Allowing 550 ft. per sec. for a Holzwarth wheel gives a velocity ratio

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[1] See, for instance, Inchley's "Theory of Heat Engines" (1920), p. 6.

[2] See "Modern Practice in Heat Engines" (1922), by Telford Petrie, pp. 130 *et seq.*

$$\alpha=\tfrac{550}{2780}=0{\cdot}198$$

Turbine efficiency curves [1] show that, for such a value, a two-row wheel is most suitable, and, moreover, that the efficiency of such a wheel at this velocity ratio is in the neighbourhood of 65 per cent. if windage, friction, and leakage are neglected.

The Holzwarth turbine uses a single two-row wheel, that is, a wheel with two rows of moving blades mounted on the rim ; and the maximum efficiency possible under such conditions is 74 per cent. with a velocity ratio of 0·29. From these values, friction, windage, and leakage have to be deducted, so that it would appear that the maximum relative efficiency possible in this design is in the neighbourhood of 71 per cent.

As far back as 1884, Sir Charles Parsons realised and pointed out that compounding was necessary in order to get high relative efficiencies in turbine work, and the gas turbine will always be at a disadvantage in this respect until some means are found to carry out expansion in a series of stages.

The above analysis could be equally well applied to the constant-pressure type of gas turbine, if it is remembered that in this case

$$U=\int V\,dp=\frac{c^2}{2g}$$

so that

$$\frac{c^2}{2g}=\frac{\gamma}{\gamma-1}RT_4\left\{1-\left(\frac{1}{X}\right)^{\frac{\gamma-1}{\gamma}}\right\}$$

and X is derived from equation (4).

Conclusion.—As was mentioned at the beginning of this chapter, it is as dangerous as it is fascinating to attempt to foretell what the future of such a subject as internal-combustion engineering may be. The whole problem is in its most interesting stage. After nearly fifty years, sufficient knowledge has been accumulated to enable clear conceptions to be formed at any rate of one branch of the subject, the reciprocator. But it must not be forgotten that the steam turbine only appeared after nearly one hundred years of the steam engine, and it is therefore by no means impossible that some form of gas turbine will eventually appear which will bear a similar relation to the internal-combustion engine that the steam turbine holds to-day with regard to its companion. Moreover, although much fundamental knowledge has been assimilated, there is far more variety in the design of interal-combustion engines than in the present-day steam engine. The battle of types at sea will probably not be settled for another ten years, whilst diversity will always be found on land, if only because of the variety of fuels available and the variety of duties for which engines are required. In conclusion, the author hopes that enough has been written to convince the would-be internal-combustion engineer that there will be plenty of scope for his ingenuity and abilities for more than one generation to come.

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[1] *Ibid.*, p. 134.

EXAMPLES

The following abbreviations are used:

C.U. = Mechanical tripos papers of Cambridge University.
L.U. = Final degree papers of London University.
M.U. = Final degree papers of Manchester University.

Note.—All pressures are in lb. per sq. in. absolute unless otherwise stated.

CHAPTER II

1. Compare, by means of sketches, the engines of Barsanti and Matteucci, as shown in patent No. 1655 of 1857, with the Otto and Langen free-piston engine of 1866.

Ans.—See Clerk, "Gas and Oil Engine" (1902), p. 12; or Robinson, "Gas and Petroleum Engine" (1890), p. 132.

2. Make a diagram of the governor and pawl and ratchet mechanism of the Otto and Langen free-piston engine as actually made.

Ans.—See Clerk, "Gas and Oil Engine" (1902), pp. 227 and 228.

3. Make a diagram of the Simon steam-gas engine of 1878 from the description on p. 16.

Ans—See Robinson, "Gas and Petroleum Engine" (1890), p. 215.

4. Show what points in the design of the Otto "Silent" engine of 1876 are still prevalent in the modern gas engine.

5. Describe, with a sketch, the governing gear of the Otto "Silent" engine of 1876. *Ans.*—See Clerk, "Gas and Oil Engine" (1902), p. 230.

6. Set out the relative positions of the working piston and of the displacer piston in a Clerk two-stroke gas engine for the four cases when each piston is in turn at the end of a stroke. Determine, also, the proportion of the in stroke travelled by the displacer piston during the period of the exhaust, if the exhaust ports are $\frac{1}{8}$th the length of the working stroke. Assume that the angle of advance of the displacer piston crank pin is 90°, and that the ratio of connecting rod to crank is 5 for the power piston and 7 for the displacer piston. *Ans.*—0·75 stroke.

7. Lay out to a quarter scale a suitable link motion of the Atkinson "cycle" engine, as tested by the judges of the Royal Society of Arts.

Note.—The proportions of Fig. 9 are taken from the small scale drawing in *Engineering*, 1886. The result does not quite agree with the stroke dimensions given in the report. It is possible, however, that the two engines were identical, and the dimensions of Fig. 9 may be scaled as a basis for this exercise. Fig. 9 is reproduced to a scale of $\frac{1}{20}$.

8. Obtain a full load indicator card and particulars of the valve settings and proportions of a gas engine in your own engineering laboratory (or elsewhere), and set out the four diagrams of Fig. 11.

9. Draw diagrams of the engine and the indicator cards, as in Fig. 11, for the Oechelhauser gas engine shown in Fig. 12.

10. Draw a diagram and explain the action of a *two-stroke* Humphrey pump described in *The Engineer*, vol. cxxi. (1916), p. 362.

Ans.—See Petrie, "Modern Practice in Heat Engines," (1922), p. 217.

11. Collect and tabulate the following data for all the gas engines mentioned in Chapter II.: Name, cycle, compression ratio, maximum pressure,

mean effective pressure, average revolutions per minute, and average horse power developed.

CHAPTER III

1. Enumerate the points of similarity and of difference in the engine shown in Daimler's 1885 patent specification (Fig. 16) and a modern petrol engine.

2. Make a diagram of a modern petrol engine on similar lines to those given in this book.

3. Write a short essay on the difficulties of classifying oil engines.

4. Obtain an indicator card and particulars of any oil engine which you have tested, and make a diagram of the engine with its valve settings.

5. Compare, with diagrams, the methods of spraying crude oil into a Diesel engine and into a solid-injection engine.

6. Show how the working substance is fired in a Diesel engine, a hot-bulb engine, and in a solid-injection engine.

7. Give a brief description of the different ways in which British shipbuilders are solving the marine oil-engine problem.

8. Describe the cycle of operations in a Diesel engine, and sketch the PV diagram corresponding to this cycle. How is the fuel injected and how is the engine governed ?

9. Describe, with sketches, any one type of solid-injection oil engine. What are the relative advantages and disadvantages of solid injection as compared with blast injection ?

CHAPTER IV

1. What is the gas constant for unit weight or mass of a gas ? Show that it can be expressed as a function of either the pressure, volume, and temperature of a perfect gas, its specific heats, or its molecular weight.

If 1 cubic ft. of hydrogen at S.T.P. weighs 0·00558 lb., and the "molecular weight" of air is 28·8, determine the specific heats at S.T.P. of air at constant pressure and at constant volume when $\gamma = 1{\cdot}404$. State the units of your answer.—M.U. *Ans.*—$K_p = 185{\cdot}2$, $K_v = 132$, ft.-lb. per lb. per °F.

2. Prove that for a perfect gas the difference between the specific heat at constant pressure and the specific heat at constant volume is the same at all temperatures and pressures.

At atmospheric temperature and pressure a cubic foot of a gas weighs 0·08 lb., and its specific heat at constant volume 0·168 B.Th.U. per lb. How much work is required to compress it to half its volume, the compression being (*a*) isothermal, (*b*) adiabatic ?—M.U. *Ans.*—(*a*) 1,468 ft.-lb. ; (*b*) 1,692 ft.-lb.

3. The characteristic constant for hydrogen is 1,382 ft.-lb. units, and its specific heat at constant pressure is 3·41. Three cu. ft. of hydrogen measured at 15 lb. per sq. in. and 18° C. are compressed adiabatically to 200 lb. per sq. in., and then expanded isothermally to the original volume of 3 cu. ft. Determine the final pressure of the gas.

Calculate the amount of heat which must be abstracted from the gas after expansion in order to reduce it to its initial state of pressure.—L.U.

Ans.—31·6 lb. per sq. in. ; 12·6 C.H.U.

4. What is meant by the expression "internal energy" as applied to the working substance of an internal-combustion engine ? Describe an experiment to show that the internal energy of a given quantity of substance depends only on its temperature. Apply this law to deduce a relation betwen the specific heats of a gas and the "gas constant."—M.U.

5. A quantity of heat dH is given to a perfect gas which undergoes expansion ; explain in words how this heat is expended, and write down an equation representing your statement.—M.U.

6. Prove that the work required to compress air adiabatically is independent of the initial pressure, and depends only on the ratio of compression.—M.U.

7. Calculate the heat received or rejected during the expansion of 1 lb. of air

from 100 lb. per sq. in. to 20 lb. per sq. in. according to the law $PV^{1·2}=\text{constant}$. If the final temperature is 32° F., what was the initial temperature?

For air $PV=53·2T$, the units being lb., ft., and °F. abs.—M.U.

Ans.—25·8 B.Th.U.; 183° F.

8. A quantity of a certain gas is compressed from initial conditions of 3 cu. ft. and 15 lb. per sq. in. to final conditions of 1·2 cu. ft. and 58 lb. per sq. in. If the specific heats of this gas at constant volume and constant pressure are 0·173 and 0·244 respectively, and the observed rise of temperature of the gas is 146° C., calculate the change in the internal energy of the gas.—L.U.

Ans.—6·17 C.H.U.

9. 40 cu. ft. of air at a temperature of 20° C. and a pressure of 30 lb. per sq. in. are compressed to a pressure of 150 lb. per sq. in. Find the amount of heat which must be taken from the air during the process if the compression is (*a*) isothermal, (*b*) according to $PV^{1·2}=\text{constant}$. Assume that $PV=96T$.—L.U.

Ans.—(*a*) 198·6 C.H.U.; (*b*) 47·2 C.H.U.

10. Deduce an expression for the rate of heat reception or rejection per unit increase of volume during the expansion of a gas. Investigate the cases when the expansion is (*a*) isothermal, (*b*) according to the general law $PV^n=\text{constant}$. In case (*b*) what inferences would you draw from your expression if n were greater than, or less than, γ? A gas expands according to the law $PV^{1·45}=\text{constant}$ ($\gamma=1·38$); find the rate of heat reception or rejection at the instant when the pressure p is 120 lb. per sq. in.—M.U.

Solution:

$$dH=\frac{PdV}{J}+C_v dT$$

$$\therefore\ \frac{dH}{dV}=\frac{P}{J}+C_v\frac{dT}{dV}$$

Also $\quad PV=RT \quad \therefore\ T=\dfrac{PV}{R}\quad$ and $\quad\dfrac{dT}{dV}=\dfrac{1}{R}\left(P+V\dfrac{dP}{dV}\right)$

∴ substituting

$$\frac{dH}{dV}=\frac{P}{J}+\frac{C_v}{R}\left(P+V\frac{dP}{dV}\right)$$

$$=\frac{P}{J}+\frac{1}{J(\gamma-1)}\left(P+V\frac{dP}{dV}\right)$$

$$=\frac{1}{J(\gamma-1)}\left(\gamma P+V\frac{dP}{dV}\right)$$

But when $\quad PV^n=C. \quad P=\dfrac{C}{V^n}\quad$ and $\quad\dfrac{dP}{dV}=\dfrac{-n\,.\,C}{V^{n+1}}=\dfrac{-nP}{V}$

∴ substituting

$$\frac{dH}{dV}=\frac{1}{J(\gamma-1)}(\gamma P-nP)$$

$$=\frac{\gamma-n}{\gamma-1}\cdot\frac{P}{J}\text{ in heat units.}$$

But $\quad n=1·45 \quad \gamma=1·38 \quad P=144\times120 \quad J=778$ or 1,400.

$$\therefore\ \frac{dH}{dV}=\frac{-0·07}{0·38}\cdot\frac{144\times120}{J}=-4·09\text{ B.Th.U.}\quad\text{or}\quad-2·27\text{ C.H.U.}$$

11. Deduce from first principles, the total amount of heat per lb. communicated to, or abstracted from, a gaseous medium under each of the following conditions:

(1) Any pressure-volume changes whatsoever;
(2) Under constant-volume conditions;
(3) Under constant-pressure conditions;
(4) Under constant-temperature conditions;
(5) Under adiabatic conditions;
(6) Under polytropic conditions, *i.e.* expansion or compression according to the general law $PV^n=C$.

Hence, show the connection between the result of (6) and the rate of heat reception or rejection

$$\frac{dH}{dV}=\frac{\gamma-n}{\gamma-1}P$$

which holds for polytropic changes.—M.U.

12. The curve of expansion on a gas-engine indicator card is given by

$$PV^{1\cdot 28}=\text{constant}$$

and the curve of compression by

$$PV^{1\cdot 35}=\text{constant}$$

At a point on the compression curve where the pressure is 15 lb. per sq. in and the volume 1700 cu. ins., the temperature is 100° C. Determine the temperature at a point:

(1) On the compression curve where the volume is 400 cu. ins.

(2) On the expansion curve where the pressure is 320 lb. per sq. in. and volume 400 cu. ins.

(3) On the expansion curve where the volume is 1700 cu. ins.

Sketch approximately the indicator diagram.—L.U.

Method.—Calculate the pressure at points (1) and (3) from the laws of compression and expansion. Then, since the mass of gas in the cylinder is assumed constant, $\frac{PV}{T}=\text{constant}$ throughout the cycle. P, V, and T are given for one point, and by substitution the values of T at the points (1), (2), and (3) can be determined. *Ans.*—(1) 346° C.; (2) 1,600° C.; (3) 977° C.

13. An internal-combustion engine has the following dimensions: dia. of cylinder, 22 ins.; stroke, 30 ins.; compression ratio, 13·5. At the end of the suction stroke the pressure is 14 lb. per sq. in. and the temperature 43° C. Compression follows the law $PV^{1\cdot 37}=C$. Determine (*a*) the pressure and temperature at the end of compression; (*b*) the weight of the charge; (*c*) the work done; and (*d*) the heat rejected during compression. Assume the specific heat at constant pressure to be 0·238, and the specific heat at constant volume to be 0·169.—L.U.

Ans.—(*a*) 483 lb. per sq. in., 557° C; (*b*) 0·471 lb.; (*c*) 63,200 ft.-lb.; (*d*) 4·1 C.H.U.

14. The cylinder of a Diesel engine is 16 ins. dia., and it has a stroke of 21 ins. During the suction stroke air is supplied at a pressure of 20 lb. per sq. in. and a temperature of 18° C. The pressure at the end of compression is 480 lb. per sq. in. The specific heat of air at constant pressure is 0·238 and at constant volume 0·169. Assuming the compression to be adiabatic, determine (1) the clearance volume of the cylinder; (2) the temperature at the end of compression.—L.U.

Ans.—(1) 0·284 cu. ft.; (2) 455° C.

15. A gas engine working on the Otto cycle has a cylinder 10 ins. diameter, and a stroke of 20 ins. The compression space is $\frac{1}{5}$ of the stroke volume. The inlet valve closes at the out-centre, and the cylinder contents then consist of air having a temperature of 40° C. and a pressure of 14·7 lb. per sq. in. The pressure reached in compression is 160 lb. per sq. in., the compression line of the indicator diagram following the curve $PV^{1\cdot 33}=\text{constant}$. Find:

(1) The mean temperature of the air at the end of compression;

(2) The temperature reached at a point in the interior of the cylinder where there has been no loss of heat.

(3) The quantity of heat lost to the cylinder walls in the course of compression.

The thermal capacity of air may be taken as 19 ft.-lb. per standard cubic foot per °C.—C.U. *Ans.*—(1) 292° C.; (2) 368° C.; (3) 0·713 C.H.U.

(2) presupposes that a central portion of the gas is compressed adiabatically.

16. The mixture in a gas engine is at a pressure of 14·7 lb. per sq. in. and temperature 150° F. at the beginning of the compression stroke. If at the end of compression the compression pressure be 120 lb. per sq. in. and the temperature 600° F., whilst the compression follows the law $PV^n=C$, find n and the compression ratio.

If at the beginning of the compression stroke a weight of pure air (equal to 40 per cent. of the weight of the total original charge) be forced into the cylinder at 60° F., what is now the temperature and pressure of the mixture at the beginning of compression?

Assuming that the compression of the modified engine starts 0·15 of the stroke after the B.D.C. with the temperature and pressure just found, what will be the new compression pressure and temperature? Also find the explosion temperature and pressure assuming that the gas supply is the same as for the original engine, and there was sufficient to give an explosion pressure double the compression pressure (both pressures being absolute).

Briefly discuss the advantages of Clerk's super-compression method for getting low flame temperatures in a gas engine.—M.U.

Solution.—

Let p_1, V_1, T_1=pressure, volume, and temperature at the beginning of compression.

,, p_2, V_2, T_2=pressure, volume, and temperature at the end of compression.

Then
$$\frac{p_2}{p_1}=\left(\frac{V_1}{V_2}\right)^n$$

and
$$\frac{T_2}{T_1}=\left(\frac{V_1}{V_2}\right)^{n-1}=\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

$$\therefore\ \frac{n-1}{n}=\log\frac{T_2}{T_1}\div\log\frac{p_2}{p_1}$$
$$=\log\frac{1060}{610}\div\log\frac{120}{14·7}$$
$$=0·263$$
$$\therefore\ n=1·36$$

Ratio of compression
$$r=\frac{V_1}{V_2}=\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}=4·68.$$

Temperature of Mixture.—40 per cent. additional weight of air at 60° F. is supplied.

$$\therefore\ \text{Temperature of mixture}=\frac{150+0·4\times60}{1·4}=124·2°\text{ F.}$$

Since this occupies the same volume, the pressure of the mixture is directly proportional to the change of weight and to the absolute temperature.

$$\therefore\ \text{Pressure}=14·7\times1·4\times\frac{460+124·2}{460+150}=19·72\text{ lb. per sq. in. abs.}$$

Volume of the mixture at the beginning of compression

$$=V_c+0·85V=V_c(1+0·85\times3·68)=V_c(1+3·13)$$

∴ New ratio of compression=4·13.

∴ New compression pressure $=4·13^{1·36}\times19·72=135·7$ lb. per sq. in. abs.

Temperature at the end of compression

$$=584·2\left(\frac{135·7}{19·72}\right)^{\frac{0·36}{1·36}}=974°\text{ F. abs.}=514°\text{ F. ord.}$$

Originally the pressure was doubled during explosion.

∴ Rise of temperature during explosion=1,060° F.

∴ Rise of temperature with 40 per cent. addition to weight of charge

$$\frac{1060}{1·4}=757°\text{ F.}$$

∴ Final explosion temperature=974+757=1731° F. abs.

∴ Final explosion pressure $=\frac{1731}{974}\times135·7=241·2$ lb. per sq. in. abs.

17. A weight of 2000 lb. falls on to the piston of a vertical air buffer cylinder. The total height the weight falls before coming to rest is 29 ft. The

initial pressure of the air in the cylinder is 30 lb. per sq. in. abs.; the diameter of the cylinder is 12 ins. The full stroke of the piston is 4 ft. (*i.e.* distance from the underside of the piston when at the top of the stroke to the inside of the cover). The index of the compression curve $n=1\cdot4$. Find the distance the piston travels after being struck by the weight and the maximum air pressure attained.—L.U.

Solution.—Work done by falling weight $=2000\times 29=58{,}000$ ft.-lb.

$\therefore$ Work done on the air in the buffer cylinder $=58{,}000$ ft.-lb.

Let $p_2=$ maximum pressure in buffer cylinder.

Let $p_1, V_1=$ initial pressure and volume.

Then work done

$$=\frac{144p_1V_1}{n-1}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right\}$$

$$\therefore\ 58{,}000=\frac{144\times 30}{0\cdot4}\times\frac{\pi}{4}\times 1^2\times 4\left\{\left(\frac{p_2}{p_1}\right)^{\frac{0\cdot4}{1\cdot4}}-1\right\}$$

$$\therefore\ \left(\frac{p_2}{p_1}\right)^{\frac{2}{7}}-1=1\cdot71$$

$$\therefore\ \frac{p_2}{p_1}=2\cdot71^{\frac{7}{2}}=32\cdot8$$

$$\therefore\ p_2=984 \text{ lb. per sq. in. abs.}$$

Volume of the air at the end of compression

$$=\left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}V_1$$

$$=\left(\frac{30}{984}\right)^{\frac{1}{1\cdot4}}3\cdot14$$

$$=0\cdot26 \text{ cu. ft.}$$

$$\therefore\ \text{Travel of piston}=4\left(\frac{3\cdot14-0\cdot26}{3\cdot14}\right)$$

$$=3\cdot67 \text{ ft.}$$

$$=3 \text{ ft. } 8 \text{ ins.}$$

18. The cylinders of a petrol motor have stroke and bore 120 and 90 mm. respectively, and a compression ratio of 4·4. The inlet valve is shut at 0·05 stroke, and the temperature and pressure of the charge is then 13·5 lb. per sq. in. and 95° C. The exhaust valve opens at 0·85 stroke and the pressure at this point is indicated as 51 lb. per sq. in. Assuming a molecular contraction of 2 per cent., find the temperature of the cylinder contents at the opening of the exhaust valve, and estimate the heat discharged in the exhaust per min. if the motor has 4 cylinders and is running at 1000 r.p.m. Take a mean specific heat of 0·24. —L.U.

Solution.—Swept or displaced volume per cylinder

$$=\frac{\pi}{4}\times 9^2\times 12=764 \text{ c.c.}$$

Clearance volume (with $r=4\cdot4$) $=\dfrac{764}{3\cdot4}=225$ c.c.

Volume of cylinder contents when inlet valve closes

$$=225+0\cdot95\times 764=951 \text{ c.c.}$$

Volume of cylinder contents when exhaust valve opens

$$=225+0\cdot85\times 764=874 \text{ c.c.}$$

But $PV=wRT$

and R has a value 2 per cent. less during the expansion stroke than during the compression stroke.

$$\therefore\ 0{\cdot}98\times\frac{13{\cdot}5\times951}{273+95}=\frac{51\times874}{\mathrm{T}}$$

where T = abs. temp. when exhaust valve opens.

$$\therefore\ \mathrm{T}=\frac{51\times874\times368}{0{\cdot}98\times13{\cdot}5\times951}$$

$$=1303^\circ\text{ C. abs.}$$

$$=1030^\circ\text{ C. ord.}$$

CHAPTERS V. AND VI

1. Prove that the ideal efficiency of an internal-combustion engine working on the Otto cycle is $1-\left(\frac{1}{r}\right)^{\gamma-1}$ where *r* is the ratio of compression. Calculate the efficiency in the case of an engine having a stroke of 16 ins., piston dia. 12 ins., and clearance volume 485 cu. ins., and find the gas consumption per I.H.P. per hour if the gas has a calorific value of 260 C.H.U. per cu. ft. and the efficiency ratio of the engine is 56 per cent.—L.U. *Ans.*—$\eta=46{\cdot}3$ per cent.; 21 cu. ft.

2. Explain the chief advantages obtained, and the difficulties to be overcome, in using a two-stroke cycle for both large and small internal-combustion engines. If the working substance may be considered to have the same properties as air, show that the efficiency E of such an engine, working on either the constant-volume or constant-pressure cycle, is expressed by

$$\mathrm{E}=1-\left(\frac{v}{\mathrm{V}+v}\right)^{\gamma-1}$$

where V is the volume of the piston displacement, v is the clearance volume, and $\mathrm{PV}^{\gamma}=$constant is the law of adiabatic compression and expansion. What is the chief error in the assumption on which the formula is based?—L.U.

3. In a gas engine working on the Otto cycle the explosive mixture enters the cylinder at atmospheric pressure and at a temperature of 115° F., and is compressed to a pressure of 85 lb. per sq. in. abs. The gas on exploding rises to a pressure of 215 lb. per sq. in. abs., and then expands to a pressure of 55 lb. per sq. in. abs. The dia. of the cylinder is 8·5 ins., stroke 18 ins., and clearance volume 0·2467 cu. ft.

If the barometric pressure is 14·7 lb. per sq. in. abs., find:

(*a*) The temperature at the beginning and at the end of expansion.

(*b*) The heat given to, or taken from, the gases during expansion.

Assume the specific heats of the mixture to be C_p, 0·2659; C_v, 0·192 before explosion; C_p, 0·2662; C_v, 0·1943 after explosion.

Which portion of the cycle do you consider the most important from an efficiency point of view?—L.U.

Ans.—(*a*) 1,950° F., 1,632° F.; (*b*) +7·75 B.Th.U.

4. One cubic foot of air and gas at an initial temperature of 63° F. (17·23° C.) and atmospheric pressure is adiabatically compressed in a cylinder to 0·4 cu. ft. Find the temperature at the end of the compression process. The gas is then burnt at constant volume, and raises the temperature of the mixture by 2,540° F. (1,411° C.). It is then adiabatically expanded to its original volume. Find the mean effective pressure on the piston, and the thermal efficiency of the engine regarded as an air engine.—L.U.

Ans.—293° F.; 90 lb. per sq. in.; 30·6 per cent.

5. A gas engine works on an ideal cycle with adiabatic compression and expansion, receiving and rejecting heat only at constant volume. Obtain the expression for its efficiency. In such an engine the piston displacement per stroke is 1 cu. ft., the clearance volume 0·2 cu. ft., and at the beginning of compression the temperature of the cylinder contents is 600° F. abs., the pressure being atmospheric. The engine receives 0·06 cu. ft. of gas per cycle (calorific value 600 B.Th.U. per cubic foot). Atmospheric pressure = 14·7 lb. per sq. in. abs. Find:

(*a*) Weight of cylinder contents.

(*b*) Pressure and temperature at the end of compression ($\gamma=1\cdot38$).

(*c*) Rise of temperature during explosion (neglect jacket loss and take $C_v=0\cdot18$).

(*d*) Pressure at the end of explosion.

(*e*) Temperature and pressure at the end of expansion.

(*f*) Efficiency of cycle.

(*g*) Efficiency of an engine working on the Carnot cycle between the same highest and lowest temperatures.—C.U.

Ans.—(*a*) 0·0797 lb.; (*b*) 180·3 lb. per sq. in.; 766° F.; (*c*) 2,510° F.; (*d*) 550 lb. per sq. in.; (*e*) 1,370° F.; 44·8 lb. per sq in.; (*f*) 51·2 per cent.; (*g*) 84 per cent.

6. The diameter of a gas-engine cylinder is 12 ins., the stroke is 20 ins., and the clearance volume is 21 per cent. of the volume swept by the piston. The temperature at the end of admission is 130° F. (54·4° C.) and the pressure 14·5 lb. per sq. in. abs. The equation to the compression curve is $PV^{1\cdot26}=$constant. The explosion supplies 420 B.Th.U. per lb. of mixture fired. The specific heat at constant pressure is 0·252, and at constant volume, 0·18. Determine:

(*a*) The maximum pressure in the cylinder.

(*b*) The maximum temperature of the mixture after firing.

(*c*) The maximum possible thermal efficiency.—L.U.

Ans.—(*a*) 463 lb. per sq. in.; (*b*) 2,804° F.; (*c*) 50·3 per cent.

7. Deduce the "air standard" efficiency from first principles, and show how far it may properly be applied to modern internal-combustion engines. Derive, using this efficiency, an expression for the mean effective pressure of the constant volume cycle.—M.U.

8. Describe the Otto or "constant volume" cycle for a gas engine, and deduce expressions for the thermal efficiency and mean effective pressure for the ideal case. The following data refer to an engine working on the ideal Otto cycle with adiabatic compression and expansion: Pressure at beginning of compression, 15 lb. per sq. in. abs.; pressure at the end of explosion, 300 lb. per sq. in. abs.; clearance volume of the cylinder, 25 per cent. of piston displacement. Find the thermal efficiency of the engine, and the mean effective pressure throughout the cycle. Take $\gamma=1\cdot4$.—M.U.

Ans.—47·5 per cent.; 46·6 lb. per sq. in.

9. A four-cylinder petrol engine, working on the Otto cycle, has cylinders $3\frac{3}{8}$ in. dia. × $4\frac{3}{8}$ in. stroke, whilst the compression ratio may be taken as 3·3. If at the end of the suction stroke the whole cylinder were filled with explosive mixture at atmospheric pressure and a temperature of 17° C., find the maximum power the engine could give at 1000 r.p.m., assuming an overall efficiency of 18 per cent. What power would actually be obtained in practice, although the efficiency might still be 18 per cent.?

1 lb. of petrol (calorific value, 19,000 B.Th.U.) requires 187 cu. ft. of air at 0° C. and 14·7 lb. per sq. in. for complete combustion.—M.U. *Ans.*—26·3 H.P.

10. Show that the ideal mean effective pressure for the constant volume cycle may be written

$$p_m=\frac{p_2 r}{(\gamma-1)(r-1)}(a-1)(r^{\gamma}-1)$$

where p_2 is the suction pressure, r is the ratio of compression, a is the explosion ratio and γ is the ratio of the specific heats.

11. In a plant in which air is taken round a reversed Joule cycle the compressing cylinder takes in air at atmospheric pressure of 14·7 lb. per sq. in., and delivers it at a pressure of 90 lb. per sq. in. At the beginning of compression the temperature is 30° F., and at the beginning of expansion it is 60° F. Determine the highest and lowest temperatures in the cycle, and the work expended per lb. of air. Take $\gamma=1\cdot4$.—C.U. *Ans.* 362° F.; −150° F.; 22,300 ft.-lb.

12. Determine the thermal efficiency of the "Joule" or "Brayton" cycle if the maximum pressure is 70 lb. per sq. in. *Ans.*—35·8 per cent.

13. Obtain an expression for the efficiency of an internal-combustion engine following the "constant pressure" or Joule cycle, in terms of the adiabatic compression ratio.

The cylinder of a Diesel engine has a bore of 6·5 ins., a stroke of 10·5 ins., and a clearance volume of 30 cu. ins. Determine the pressure of the air at the end of the adiabatic compression stroke, if it follows the law $pv^{1\cdot 3}$=constant.

Also determine the theoretical efficiency of the engine.—C.U.

Ans.—397 lb. per sq. in., 53·3 per cent. (on the modified Joule cycle with constant-volume exhaust).

14. In an ideal air engine receiving heat at constant pressure and rejecting heat at constant volume (C_p 0·2375, C_v 0·169), the expansion and compression are adiabatic, and the ratio of compression is 15, and of expansion 7. The temperature at the beginning of compression is 104° F. The pressure at the beginning of compression is 15 lb. per sq. in., and at release 45 lb. per sq. in. Find the work done per lb. of working substance.—L.U.

Ans.—268·5 B.Th.U. or 208,700 ft.-lb.

15. Find an expression for the ideal efficiency of the Diesel cycle, and calculate this efficiency in the case of an engine having a compression ratio of 13·5, and in which the fuel is cut off at 0·05 of the stroke.—L.U. *Ans.*—60·7 per cent.

16. If the compression and expansion ratio of an engine working on the Diesel cycle are denoted by r_c and r_e respectively, and the ratio of the specific heats is denoted by γ, show that the ideal efficiency is

$$1-\frac{1}{\gamma}\left\{\frac{\left(\frac{r_c}{r_e}\right)^{\gamma}-1}{r_c^{\gamma-1}\left(\frac{r_c}{r_e}-1\right)}\right\}$$

A Diesel engine develops 165 B.H.P. and has an efficiency ratio of 0·55 on the brake. If r_c=13·8, r_e=7·4, and the lower calorific value of the oil is 19,000 B.Th.U. (10,500 C.H.U.) per lb., find the consumption of oil in lb. per B.H.P. per hour. —L.U. *Ans.*—0·41 lb. per B.H.P. per hour.

17. Describe the principle of action of an engine following the Diesel cycle, and account for its high efficiency. The cylinder of such an engine has a diameter of 16 ins., and a stroke of 22 ins., and it is supplied with compressed air at 5 lb. per sq. in. above atmosphere. Find the clearance required to give a compression pressure of 520 lb. per sq. in. at the end of compression if the law of adiabatic compression is $PV^{1\cdot 4}$=constant. Also find the temperature at the end of compression if the initial temperature is 20° C. (68° F.).—L.U.

Ans.—473 cu. ins.; 474° C.

18. Calculate the mean effective pressure of a Diesel engine card for a compression ratio of 14, a maximum pressure of 480 lb. per sq. in., and a cut-off ratio of 2·3. Assume compression and expansion to follow the same law of PV^n=constant, with constant specific heats. The suction pressure is 14 lb. per sq. in. What would be the diameter of the cylinder of such an engine, working on the four-stroke cycle, to develop 125 I.H.P. with a piston speed of 800 ft. per min.?—M.U.

Ans.—97·8 lb. per sq. in.; 16·4 ins.

19. Find the mean effective pressure, efficiency, H.P. developed in the cylinder, and the principal temperatures and pressures during the cycle, in the case of a four-stroke Diesel engine, given the following data:

Diameter of cylinder, 12 ins.; stroke, 18 ins.; speed, 200 r.p.m.; ratio of compression, 13; cut-off ratio, 2; temperature at the beginning of compression, 150° F.; whilst the pressure at this instant is 14·7 lb. per sq. in.

If the fuel has a calorific value of 18,000 B.Th.U. per lb. calculate the oil consumption per hour, the efficiency ratio of the engine being 56 per cent.—M.U.

Ans.—η, 58·1 per cent.; p_m, 90·5 lb. per sq. in.; I.H.P., 46·6.

Oil consumption per hour:

$$\text{Weight of cylinder contents per cycle}=\frac{\pi}{4}\times\frac{13\times 1^2\times 1\cdot 5\times 14\cdot 7\times 144}{12\times 53\cdot 2(460+150)}=0\cdot 0834\text{ lb.}$$

The temperature at the end of compression=1700° F. abs. Since the cut-off ratio is 2, the rise of temperature during injection of the oil will also be 1700° F.

∴ Taking $C_p=0{\cdot}237$, heat supplied during combustion

$$=0{\cdot}237\times 0{\cdot}0834\times 1700$$
$$=33{\cdot}6 \text{ B.Th.U.}$$

∴ Oil used per cycle $=\frac{33{\cdot}6}{18000}$ lb. $=0{\cdot}00186$ lb.

$$\therefore \text{Oil used per hour}=0{\cdot}00186\times\frac{200}{2}\times 60=11{\cdot}2 \text{ lbs.}$$

With efficiency ratio 56 per cent., oil used per hour $=\frac{11{\cdot}2}{0{\cdot}56}=20$ lbs.

20. The following data refer to a Diesel engine working on the four-stroke cycle. Working volume of cylinder, 3 cu. ft.; clearance volume, 0·25 cu. ft.; pressure at the beginning of compression, 14·7 lb. per sq. in.; temperature of the air at the beginning of compression, 100° F.; r.p.m., 200; oil used per min., 0·55 lb.; calorific value of the oil, 18,000 B.Th.U. per lb. Calculate the thermal efficiency, mean effective pressure on the piston, and the indicated horse power of the engine, assuming that the working substance is air only, and of constant mass throughout the cycle. Take for air, $R=53{\cdot}2$; C_p 0·237; $\gamma=1{\cdot}41$. —M.U. *Ans.*—58·4 per cent.; 104·0 lb. per sq. in.; 136·1 I.H.P.

21. Calculate the thermal efficiency, work done per cubic foot of piston displacement, and the mean pressure in a Diesel engine without heat losses, whose clearance volume is 7½ per cent. of the total cylinder volume. Assume the temperature at the beginning of the compression stroke to be 160° F., the working substance a perfect gas with the same properties as air. Heating value 420 B.Th.U. per lb. of stuff.

Calculate also the same properties when the cycle of operations is the true constant pressure cycle.—M.U.

Ans.—Diesel: 58·3 per cent.; 13,370 ft.-lb.; 92·8 lb. per sq. in.
Joule, 64·5 per cent.; 7,060 ft.-lb.; 49·0 lb. per sq. in.

22. A "Sprayer" oil engine with a compression ratio of 10, operates on the dual-combustion cycle. One pound of working substance, assumed to be air, is drawn in at 373° C. abs. and 14·0 lb. per sq. in. The ratio of maximum absolute pressure to that at the end of compression is found to be 1·5, and the cut-off ratio is 1·3. Determine the maximum temperature of the cycle, and compare the theoretical efficiency of the cycle with the air standard efficiency for the same compression ratio.—M.U.

Ans.—1555° C.; 58·8 per cent.; air-standard efficiency, 60·1 per cent.

23. Deduce expressions for the thermal efficiency and mean effective pressure of an engine working on the dual-combustion cycle with incomplete expansion. Calculate the thermal efficiency and m.e.p. of a "Sprayer" oil engine, having a compression ratio of 10, a maximum allowable pressure of 550 lb. per sq. in. when using crude oil having a lower calorific value of 18,000 B.Th.U. per lb. and an air-to-fuel ratio of 36. Make suitable assumptions for any data not given.—M.U.

Ans.—m.e.p. 93·6 lb. per sq. in.; 57·7 per cent.

24. Deduce the ideal thermal efficiency of the following two thermodynamic cycles A and B.

Cycle A (Sequence of Events):

(*a*) Adiabatic compression of working mixture from pressure p and volume v.
(*b*) Heat supplied at constant volume.
(*c*) Heat supplied at constant pressure.
(*d*) Adiabatic expansion to volume v.
(*e*) Heat rejection at constant volume until initial pressure p is attained.

Cycle B:

(*a*), (*b*), and (*c*) events the same as for cycle A.
(*d*) Adiabatic expansion to pressure p.
(*e*) Heat rejection at constant pressure until initial volume v is attained.

Hence, show that the ideal thermal efficiencies of the constant volume, constant pressure, Diesel engine, Blackstone engine, and Humphrey pump cycles are particular cases of cycles A or B.—M.U.

Ans.—Cycle A :

$$1-\frac{\alpha\rho^{\gamma}-1}{r^{\gamma-1}\{(\alpha-1)+\gamma\alpha(\rho-1)\}}\text{ ; Cycle B : }1-\frac{\gamma(\rho\alpha^{\frac{1}{\gamma}}-1)}{r^{\gamma-1}\{(\alpha-1)+\gamma\alpha(\rho-1)\}}$$

25. Assuming constant specific heat of the working medium, determine the compression ratio giving maximum thermal efficiency for the following cycle of events :

(*a*) Adiabatic compression from volume v_1 and pressure 14 lb. per sq. in. to a volume of 1·5 cu. ft. and pressure p_2.
(*b*) Addition of heat at constant volume from pressure p_2 to pressure p_3.
(*c*) Addition of heat at constant pressure from volume v_2 (=1·5 cu. ft.) to a volume v_4(=3·75 cu. ft.).
(*d*) Adiabatic expansion from volume v_4 and pressure p_3 to a volume v_5(=18 cu. ft.) and pressure p_5.
(*e*) Rejection of heat at constant volume from pressure p_5 to pressure p_6 (=14 lb. per sq. in.).
(*f*) Rejection of heat at constant pressure from volume v_5 to volume v_1.

Assume $p_3v_3^{\gamma}=p_6v_5^{\gamma}$ and $\gamma=1·4$.—M.U. *Ans.*—7.

For method of solution see *Engineering*, April 9, 1920, p. 467.

26. Determine the ideal efficiency of the Atkinson engine used in the trials of 1888, when the maximum pressure was 180 lb. per sq. in. and the compression pressure 50 lb. per sq. in.

Compare this efficiency with the corresponding constant volume efficiency.

Ans.—44·8 per cent. ; 37·5 per cent.

27. Explain the cycle involved in the four-stroke Humphrey gas pump.

Compare the thermal efficiency of the cycle on which the Humphrey pump works with the air-standard efficiency.

28. Describe the cycle of operations of a Humphrey pump, illustrating the same by a diagrammatic representation of the main pump features.

Deduce a formula for the thermal efficiency of the pump in terms of the compression and explosion ratios, and also a formula for the mean effective pressure of the pump, in terms of the thermal efficiency, the compression and explosion ratios, and the pressure at the end of compression. Assume constant specific heats of the working substance.—M.U.

Ans.—See text for solutions.

29. In an engine following the Diesel cycle the temperatures at the beginning of compression and at the end of combustion are T_2 and T_4 °C. abs. respectively. Show that the area of the indicator diagram will be a maximum when the ratio of compression is given by

$$r=\left(\frac{T_4}{T_2}\right)^{\frac{\gamma}{(\gamma-1)(\gamma+1)}}$$

Ans.—See Chapter XII. for method of solution.

30. The suction temperature in a Diesel engine is 300° C. abs., and the maximum temperature is limited to 1800° C. abs. Find the ratio of compression required for maximum work done. In these circumstances, what must be the calorific value of the working substance ? *Ans.*—11·7 ; 36 B.Th.U. per cu. ft.

31. A Humphrey pump uses a mixture having a calorific value of 40 B.Th.U. per cu. ft. If the temperatures at the beginning of compression and the end of explosion are 373° C. abs. and 2000° C. abs., what ratio of compression will be required ? Calculate the thermal efficiency and mean effective pressure, assuming $\gamma=1·4$, $c_v=0·169$. *Ans.*—6·0 ; 58·4 per cent. ; 39·2 lb. per sq. in.

CHAPTER VII

1. When a gaseous mixture is exploded in a closed vessel, the maximum pressure attained falls considerably short of that which might have been expected from calculation. Discuss briefly three hypotheses which have been put forward to explain this. Give an account of any one set of experiments which have been

carried out in connection with this subject, and show what bearing the phenomenon has on the behaviour of the working substance in an internal-combustion engine. —M.U.

2. Describe some experiments which have been carried out on the internal energy of gases at high temperatures. How does the increasing specific heat of the gases at high temperatures affect the efficiencies which are theoretically attainable in an internal combustion engine ?—M.U.

3. Assuming for a gaseous mixture that $C_v=0{\cdot}194+0{\cdot}000051t$, t being the temperature in °C., and taking R as 99·4 ft.-lb. per lb. per °C., show by a graph how γ varies with t. Between what limits does γ always lie ?—M.U.

4. A mixture of one volume of coal gas and nine volumes of air is exploded in a closed vessel at a temperature of 100° C. The lower calorific value of the coal gas is 550 B.Th.U. per cu. ft. If T is the absolute temperature in °C., the instantaneous specific heat at constant volume of the products of combustion is $16{\cdot}8+0{\cdot}008T$ in ft.-lb. per cu. ft. per °C. What will be the temperature reached if there is no dissociation and no heat loss ? What is the ratio of the specific heats at the maximum temperature ? Take $R=99$ ft.-lb. per lb. per °C.—M.U.

Ans.—2000° C. abs. ; $\gamma=1{\cdot}235$.

5. The curve of internal energy of a gas engine mixture published by the Gaseous Explosions Committee of the British Association is shown by Sir Alfred Ewing to conform to the formula

$$E=5{\cdot}2t+0{\cdot}00043t^2+0{\cdot}0000002t^3$$

where t is in degrees Centigrade and the units are gramme-calories per mol.

Obtain an expression for the corresponding specific heat at constant volume in terms of the temperature expressed in °C. abs.

If 3·9 gm.-cals. per mol. equal 1 ft.-lb. per standard cu. ft., and 1 cu. ft. of this mixture weighs 0·07833 lb., convert the expression for the specific heat into ft.-lb. per lb. units.—M.U.

Ans.—$5{\cdot}01+0{\cdot}000532T+0{\cdot}6\times10^{-6}T^2$ molecular heat units °C. (abs.).
$249{\cdot}6+0{\cdot}0265T+29{\cdot}9\times10^{-6}T^2$ ft.-lb. per lb. in °C. (abs.).

6. Deduce a general expression for the rate of heat reception or rejection per unit volume of a gas, assuming that the specific heats are given by $K_p=A+ST$; $K_v=B+ST$, where A, B, and S are constants, and T is the temperature in °C. abs. Thence obtain the particular expression corresponding to expansion or compression of the gas according to the law $PV^n=C$.

Ans.—See text for solutions.

7. When the specific heats of a gaseous medium are assumed to vary as follows :

$$C_p=A+ST \qquad\qquad C_v=B+ST$$

where T=temperature in deg. C. abs., and A, B, and S are constants, show that the equation for adiabatic expansion may be written

$$P^B V^A e^{ST}=\text{constant}$$

where e is the base of hyperbolic logarithms and P and V are the pressure and volume of the medium.—M.U.

8. The specific heat of a gas at constant volume may approximately be expressed as a function of the temperature or

$$C_v=a+bT$$

where a and b are constants, and T is the absolute temperature °C. If a is 0·124 and b 0·00005 in heat units, and the value of R in ft.-lb. units in the equation $PV=RT$ for this gas is 94, determine the specific heat at constant pressure. Also show that if the gas expands adiabatically from a temperature T_2 to a temperature T_1, the work done is :

$$W=0{\cdot}124(T_2-T_1)+0{\cdot}000025(T_2^2-T_1^2)$$ —L.U.

Ans.—$C_p=0{\cdot}191+0{\cdot}00005T$.

9. Show that for an adiabatic compression or expansion with variable specific heats, *i.e.* $C_p=A+ST$; $C_v=B+ST$, the following equation gives the relation

between the ratio of compression r, and the highest and lowest absolute temperatures T_2 and T_1 respectively:

$$\log_e r = \frac{B}{A-B}\log_e\frac{T_2}{T_1} + \frac{S}{A-B}(T_2-T_1)$$

Solution.—For an adiabatic with variable specific heats:

$$P^B V^A e^{ST} = \text{constant}$$

$$\therefore\ P_1{}^B V_1{}^A e^{ST_1} = P_2{}^B V_2{}^A e^{ST_2}$$

$$\therefore\ \left(\frac{P_1V_1}{P_2V_2}\right)^B = e^{S(T_2-T_1)}\left(\frac{V_2}{V_1}\right)^{A-B}$$

$$\therefore\ \left(\frac{V_1}{V_2}\right)^{A-B} = \left(\frac{P_2V_2}{P_1V_1}\right)^B e^{S(T_2-T_1)}$$

But $\frac{V_1}{V_2}$ = ratio of compression or expansion $= r$

and
$$\frac{P_2V_2}{P_1V_1} = \frac{T_2}{T_1}$$

$$\therefore\ r^{A-B} = \left(\frac{T_2}{T_1}\right)^B e^{S(T_2-T_1)}$$

Taking logs $$\log_e r = \frac{B}{A-B}\log_e\frac{T_2}{T_1} + \frac{S}{A-B}(T_2-T_1)$$

10. Using the equation given in the previous question, and taking $A=0·242$, $B=0·171$, and $S=0·0000411$, plot curves showing the variation of the temperature (T) and pressure (p) with the ratio of compression (r). Assume the initial temperature and pressure to be 100° C. and 14 lb. per sq. in. respectively.

Ans.—Calculate values of r corresponding to different values of the final temperature T_2 Then use the equation $\frac{PV}{T}$ = constant, to determine the compression pressures.

Ans.—r	2	4	6	8	10	12	14	16
p	36·4	93·7	162·2	239·5	323·5	414	508	608
T	485	624	721	798	862	918	967	1012

11. If the compression is adiabatic with variable specific heats, and the initial temperature and pressure as also the values of A, B, and S are as given in question 10, calculate the temperature and pressure at the end of compression, and the work done in compressing one cubic foot of the mixture in the following cases:

(*a*) Otto cycle gas engine with $r=5·5$;
(*b*) Solid injection oil engine with $r=10$;
(*c*) Diesel oil engine with $r=14$.

Ans.—(*a*) $T=700$; $p=146$; $W=6,915$ ft.-lb.
(*b*) $T=862$; $p=323$; $W=10,520$ ft.-lb.
(*c*) $T=967$; $p=508$; $W=12,920$ ft.-lb.

12. Draw the internal-energy curve and the constant-pressure energy curve for one standard cubic foot of a gas or oil engine mixture. The scale of temperature must read up to 3000° C. abs., and the specific heats may be taken as in question 10.

Ans.—T° C. abs. . . .	500	1,000	1,500	2,000	2,500	3,000
E_v ft.-lb. per S.C.F.	2,630	13,670	25,870	39,130	53,600	69,130
E_p ft.-lb. per S.C.F.	3,618	18,550	34,640	51,770	70,130	89,560

13. In a gas engine working on the constant-volume cycle with a compression ratio of 5·5 to 1, 18 cu. ft. of working substance at 14 lb. per sq. in. and 100° C. are compressed to 135 lb. per sq. in. Find the increase of internal energy due to compression when $K_v=220+0·0623T$, in ft.-lb. per lb. per °C. abs.

If the lower calorific value of the gas is 550 B.Th.U. per cubic ft., and the ratio of air to fuel is 11·3 to 1, what is the total internal energy from absolute zero at the end of combustion?—M.U. *Ans.*—782,200 ft.-lb.

14. Plot the adiabatic expansion curve with variable specific heats for an Otto cycle gas engine, the pressure, volume, and temperature at the beginning of expansion being 400 lb. per sq. in., 0·30 cu. ft., and 2000° C. abs. respectively, and the volume at the end of expansion being 2·16 cu. ft. Use the data for the specific heats given in question 10.

15. Draw pressure-volume diagrams illustrating the Diesel engine cycle, and the dual-combustion cycle as approximately followed by some modern solid injection oil engines. Indicate the final compression pressures and maximum pressures commonly reached in these two types.

A solid-injection oil engine has a compression ratio of 8·5, a final compression pressure of 300 lb. per sq. in., the cut-off ratio at full load being 1·2. The pressure and temperature at the beginning of compression are 14 lb. per sq. in., and 100° C. respectively. Estimate the probable fuel consumption in lb. of oil per I.H.P. hour, assuming the oil supplied to have a calorific value of 10,000 C.H.U. per lb.

Take : $K_p = 317 + 0·0623$ T. ft.-lb. per lb. per °C.
$K_v = 220 + 0·0623$ T. ft.-lb. per lb. per °C.

T being in °C abs.—M.U. *Ans.*—0·419 lb.

16. A gas engine working on the Otto cycle has a cylinder 12 ins. dia. and 20 ins. stroke, and the clearance volume is 20 per cent. of the volume displaced by the piston. The temperature and pressure at the end of the suction stroke may be assumed to be 100° C. and 14 lb. per sq. in. respectively. If the mixture supplied has a calorific value of 420 B.Th.U. per lb., and has specific heats as given in question 10, calculate the thermal efficiency of the engine and the mean effective pressure on the piston. *Ans.*—The ratio of compression is 6.

From question 10, we find that when $r=6$, $p=162$ lb. per sq. in. and T=722° C. abs.

From the curves of question 12 :
Internal energy per standard cu. foot at 722° C.=7,250 ft.-lb.
Vol. of 1 lb. of mixture at S.T.P.=12·77 cu. ft.
∴ Calorific value of mixture per standard cubic foot

$$= \frac{420}{12·77} \text{ B.Th.U.} = \frac{420}{12·77} \times 778 = 25{,}550 \text{ ft.-lb.}$$

Total internal energy per standard cubic foot at the end of explosion
=7,250+25,550 ft.-lb.
=32,800 ft.-lb.

∴ Temperature at the end of explosion=1,766° C. abs.
=1,493° C. ord.

The temperature at the end of expansion is best found by trial from the equation

$$\log_e r = \frac{B}{A-B} \log_e \frac{T_3}{T_4} + \frac{S}{A-B}(T_3 - T_4)$$

Substituting the known values

$$1·7918 = 2·41 \log_e \frac{1766}{T_4} + \frac{1766 - T_4}{1728}$$

from which by trial T_4, the temperature at the end of expansion=1004° C. abs. =731° C. ord.

From the energy curve, the internal energy at temperature T_4=13,770 ft.-lb. per standard cubic foot.

∴ Heat rejected from T_4 to $T_1 = H_2 = 13{,}770$ ft.-lb.
and heat supplied from T_2 to $T_3 = H_1 = 25{,}550$ ft.-lb.
∴ Work done per standard cubic ft. $= H_1 - H_2 = 11{,}780$ ft.-lb.

$$\therefore \text{ Thermal efficiency} = \frac{\text{work done}}{\text{heat supplied}} = \frac{11780}{25550}$$
$$= 0·462 \text{ or } 46·2 \text{ per cent.}$$

Let p_m = mean effective pressure in lb. per sq. in.
and V = volume displaced by piston

$$= V_1 - V_2 = V_1\left(1 - \frac{1}{r}\right) = \frac{5}{6} V_1.$$

At temperature 100° C. and pressure 14 lb. per sq. in., 1 standard cubic foot occupies a volume of 1·434 cu. ft.

$$\therefore\ V_1=1{\cdot}434.$$

$$\therefore\ \text{Work done}=144p_mV=11780\ \text{ft.-lb.}$$

$$\therefore\ p_m=\frac{11780\times6}{144\times5\times1{\cdot}434}$$

$$=68{\cdot}4\ \text{lb. per sq. in.}$$

17. A gas engine working on the Otto cycle uses 0·0717 cu. ft. of gas per explosion, the calorific value, and the temperature and pressure of the gas being 395 B.Th.U. per cu. ft., 43° C. and 29·86 ins. Hg respectively. The cylinder dimensions are: diameter, 9 ins.; stroke, 17 ins.; the ratio of compression is 5·24, and the r.p.m. 250. Assuming the mixture in the cylinder at the end of the suction stroke to have a temperature of 100° C. and a pressure of 14 lb. per sq. in., calculate the thermal efficiency, and the I.H,P. of the engine.

Ans.—41·8 per cent.; 34·9 I.H.P.

18. A Diesel engine running at 150 r.p.m. has a ratio of compression of 13. It uses 1·05 lb. of oil per minute, the calorific value of the oil being 20,000 B.Th.U. per lb. The cylinder dimensions are: diameter, 22 ins.; stroke, 30 ins.; and the temperature and pressure at the beginning of compression are 100° C. and 14 lb. per sq. in. respectively. Calculate the thermal efficiency and the I.H.P. of the engine. *Ans.*—48·7 per cent.; 241 I.H.P.

CHAPTER VIII

1. Define entropy, and explain how it is measured. Deduce an expression for the change of entropy which occurs in 1 lb. of gas expanding according to the law PV^n=constant. Assume constant specific heats.

2. Calculate the change of entropy produced in 1 lb. of gas expanding according to the law $PV^{1\cdot3}$=constant; the initial pressure being 200 lb. per sq. in., and temperature 1800° C. abs., and the final pressure 50 lb. per sq. in.

Assume $C_p=0{\cdot}237$; $C_v=0{\cdot}172$. *Ans.*—0·0141.

3. In the case of a perfect gas evaluate the definite integral $\int\frac{dH}{T}$ between any two states defined by v_1T_1 and v_2T_2, where dH is the heat received per lb. of the gas at absolute temperature T, irreversible processes being excluded.

Also if the relation between pressure and volume is given by the equation PV^n=constant, show that the total heat received is proportional to T_2-T_1, and find the complete expression for it.—C.U.

4. During the expansion stroke of a gas engine the gas expands from pressure 400 lb. per sq. in., and volume 0·2 cu. ft., to pressure 40 lb. per sq. in., and volume 1·1 cu. ft. according to the law PV^n=constant. Find the value of the index n, and calculate the change of entropy if the temperature at the beginning of expansion is 2000° C. abs.

Assume $\gamma=1{\cdot}38$; $C_v=0{\cdot}170$. *Ans.*—1·35; 0·00868.

5. Plot the $T-\phi$ curves for 1 lb. of a gaseous mixture heated from a temperature of 300° C. abs. to a temperature of 2000° C. abs.: (*a*) at constant volume, (*b*) at constant pressure.

Assume $C_p=0{\cdot}237$; $C_v=0{\cdot}172$.

6. The following particulars relate to a gas engine working on the Otto cycle: Diameter of cylinder, 9 ins.; stroke, 17 ins.; clearance volume, 255 cu. ins. An indicator diagram was taken from the engine during a trial, and the pressures on the expansion and compression curves measured at intervals of one-fifth of the stroke. The results are entered in the table below.

Ordinate	0	1	2	3	4	5
Pressure (exp. curve) .	50	62·5	82	114	176	334
Pressure (comp. curve) .	15	18	23	32	52	116

If the temperature at the beginning of compression is assumed to be 100° C.; C_p, 0·237; C_v, 0·169, draw the $T-\phi$ diagram.

7. Show, from first principles, that for unit mass of a gas, the amount of heat dH supplied from without, in a change from P, V, T, to $P+dP$, $V+dV$, $T+dT$, is given by

$$dH=K_v dT+P dV=K_v dT-V dP$$
$$=\frac{1}{\gamma-1}(\gamma P dV+V dP)$$

where $\gamma=K_p/K_v$.

State distinctly why K_p and K_v depend only on T. Also show that if $K_v=a_1+bT$, $K_p=a_2+bT$, where a_1, a_2, and b are constants, the change of entropy per unit mass in passing from the state (V_1, T_1) to the state (V_2, T_2) is given by

$$\phi_2-\phi_1=a_1 \log_e \frac{T_2}{T_1}+b(T_2-T_1)+R \log_e \frac{V_2}{V_1}$$

where $R=a_2-a_1$.—M.U.

8. Plot the $T-\phi$ curves for 1 lb. of a gaseous mixture heated from 300° C. abs. to 2000° C. abs. : (*a*) at constant volume ; (*b*) at constant pressure.

Assume $C_p=0{\cdot}242+0{\cdot}0000411T$; $C_v=0{\cdot}171+0{\cdot}0000411T$.

9. Deduce an expression for the change of entropy of a gas expanding according to the law $PV^n=$constant, the specific heats being assumed to vary according to a linear law with the absolute temperature. Hence calculate the change of entropy of 1 lb. of gas expanding from pressure 350 lb. per sq. in. and temperature 1800° C., to a pressure of 40 lb. per sq. in., when the index $n=1{\cdot}3$.

$$\text{Assume } C_p=0{\cdot}223+0{\cdot}075\frac{T}{1000}; \quad C_v=0{\cdot}152+0{\cdot}075\frac{T}{1000}$$

Ans.—0·095.

10. Using the particulars given in question 6, draw the $T-\phi$ diagram when the specific heats of the working mixture are as follows :

$$C_p=0{\cdot}242+0{\cdot}0411\frac{T}{1000}; \quad C_v=0{\cdot}171+0{\cdot}0411\frac{T}{1000}$$

11. Discuss the merits of the $T-\phi$ diagram in studying the heat exchanges taking place during an internal-combustion engine cycle.

12. Compare the actual $T-\phi$ diagram obtained for a given engine, with the theoretical $T-\phi$ diagram for the same cycle of operations. Discuss the main points of difference between the two diagrams.

CHAPTER IX

1. Explain the difference between the higher and lower calorific values of a fuel gas.

The gas from a suction-gas plant gave the following analysis by volume :

CH_4, 0·65 per cent.; CO_2, 6·57 per cent. ; H_2, 18·73 per cent. ;
CO, 25·07 per cent. ; N_2, 48·98 per cent.

Find the volume of air required for the complete combustion of 100 cu. ft. of the gas, and the lower calorific value of the gas per standard cubic foot, having given that

CO has a calorific value of 4,370 B.Th.U. per lb.
H_2 has a calorific value of 61,260 B.Th.U. per lb.
CH_4 has a calorific value of 26,400 B.Th.U. per lb.

The weight of one standard cubic foot of hydrogen may be taken as 0·0056 lb. Air contains 21 per cent. of its volume of oxygen. The latent heat of steam may be taken as 900 B.Th.U. per lb.—C.U.

Ans.—110·4 cu. ft. ; 148·8 B.Th.U. per cu. ft.

2. A coal gas gives the following volumetric analysis : H_2, 0·48 ; CH_4, 0·31 ; CO, 0·110 ; N_2, 0·020 ; CO_2, 0·035 ; C_2H_4, 0·025 ; O_2, 0·020. Calculate the higher calorific value of this gas employing the usual carbon and hydrogen figures, viz. : 1 lb. of carbon burning to CO evolves 2,420 C.H.U., and burning to CO_2, 8,080 C.H.U. ; 1 lb. of hydrogen evolves 34,000 C.H.U.

State the assumptions made in such a calculation. Compare the result with the value obtained when the following calorific values for the different constituents are used. Calorific values: C.H.U. per cu. ft.: H_2, 191; CH_4, 575; CO, 188; C_2H_4, 930.—L.U.

Ans.—

Constituent.	Volume of constituent in 1 cu. ft. of gas.	Molecular weight.	Relative weights in gas.
H_2	0·480	2	0·480
CH_4	0·310	16	$0{\cdot}31\times\frac{16}{2}=2{\cdot}480$
CO	0·110	28	$0{\cdot}11\times\frac{28}{2}=1{\cdot}540$
N_2	0·020	28	$0{\cdot}02\times\frac{28}{2}=0.280$
CO_2	0·035	44	$0{\cdot}035\times\frac{44}{2}=0{\cdot}770$
C_2H_4	0·025	28	$0{\cdot}025\times\frac{28}{2}=0{\cdot}350$
O_2	0·020	32	$0{\cdot}020\times\frac{32}{2}=0{\cdot}320$
			6·220

The density of the coal gas $=6{\cdot}220\times0{\cdot}005591$
0·03478 lb. per cu. ft.

where density of hydrogen $=0{\cdot}005591$ lb. per cu. ft.

The weight of carbon in the gas burning to CO_2 is represented by the sum of the weights of the carbon in the CH_4 and C_2H_4

$=\{\frac{12}{16}\times2{\cdot}48+\frac{24}{28}\times0{\cdot}35\}0{\cdot}05591$ lb. per cu. ft.
$=0{\cdot}01208$ lb. per cu. ft.

The weight of carbon in the gas burning from CO to CO_2

=weight of carbon in the CO
$=\frac{12}{28}\times1{\cdot}54\times0{\cdot}005591$ lb. per cu. ft.
$=0{\cdot}003690$ lb. per cu. ft.

The weight of hydrogen in the gas

=wt. of free hydrogen+wt. of hydrogen in the CH_4 and C_2H_4
$=(0{\cdot}48+\frac{4}{16}\times2{\cdot}48+\frac{4}{28}\times0{\cdot}35)0{\cdot}005591$
$=0{\cdot}006430$ lb. per cu. ft.

∴ Calorific value of the gas per cu. ft.

$=0{\cdot}01208\times8080+0{\cdot}00369\times(8080-2420)+0{\cdot}00643\times34{,}000$
$=97{\cdot}61+20{\cdot}88+217{\cdot}62$
$=336{\cdot}1$ C.H.U.

Calorific value calculated from the constituent gases:

H_2	191	0·48	$0{\cdot}48\times191=91{\cdot}7$
CH_4	575	0·31	$0{\cdot}31\times575=178{\cdot}3$
CO	188	0·11	$0{\cdot}11\times188=20{\cdot}7$
C_2H_4	930	0·025	$0{\cdot}025\times930=23{\cdot}2$
			313·9

∴ Calorific value=313·9 C.H.U. per cu. ft

The second method gives a lower, and at the same time a more correct result; in general, a portion of the heat of combustion is absorbed in splitting up any hydrocarbon into its component elements.

3. During a gas engine trial a volumetric analysis of the gas was found to be:
Hydrogen, 26 per cent.; carbonic oxide, 12 per cent.; carbonic acid, 14 per cent.; marsh gas, 2 per cent.; nitrogen, 46 per cent.

Calculate the volume of air required per cubic foot of gas for complete combustion.—M.U. *Ans.*—1·095 cu. ft.

4. The gas obtained from a producer is found to have the following composition by volume:
Hydrogen, 19·3 per cent.; marsh gas, 1·2 per cent.; carbon dioxide, 9·4 per cent.; carbon monoxide, 20·1 per cent.; nitrogen, 50 per cent.

Calculate the volume of air required and the calorific value of the gas.—M.U
Ans.—1·052 cu. ft.; L.C.V. 75·9 C.H.U. per cu. ft.

5. Calculate the volume of air which is necessary and sufficient to burn one cubic foot of gas having the following volumetric percentage composition: H_2, 40·1; CH_4, 28·6; CO, 12·7; O_2, 0·8; CO_2, 4·1; N_2, 13·7.

A gas engine running at constant speed and firing every cycle had the mixture strength varied and the following results were obtained:

Ratio air to gas	11·7	9·13	7·74	5·38	4·40	3·14
I.H.P.	2·96	6·20	7·90	8·84	9·06	8·62
Gas consumption per min. cu. ft.	1·57	2·11	2·50	3·19	3·75	4·95

Taking the calorific value of the gas as 282 C.H.U. per cu. ft., draw a graph showing the variation of thermal efficiency with mixture strength.—L.U.

Ans.—Air required, 3·98 cu. ft.
Thermal efficiency, 15·7, 24·5, 26·4, 23·1, 20·2, 14·5.

6. A gas engine uses producer gas which has a volumetric percentage analysis of CO, 11·0; H_2, 29·0; CH_4, 1·8; CO_2, 16·1; N_2, 42·1. Calculate the volume of air required to completely burn one cubic foot of this gas.

If the air supplied to the gas engine be 50 per cent. in excess of this quantity, what is the calorific value of one cubic foot of cylinder mixture. The calorific values of CO, H_2, and CH_4 are 190, 162, and 535 C.H.U. per cubic foot respectively. What is the gas consumption per H.P. per hour if the engine efficiency is 23·6 per cent.? —L.U. *Ans.*—1·123 cu. ft.; 28·9 C.H.U.; 77·2 cu. ft.

7. The volume analysis of a producer gas is: H_2, 14 per cent.; CH_4, 2 per cent.; CO, 22 per cent.; CO_2, 5 per cent.; O_2, 2 per cent.; N_2, 55 per cent. Find the air required for the perfect combustion of one cubic foot of this gas. If 40 per cent. excess air is supplied find the volume analysis of the dry products of combustion. Air contains O_2, 20·9 per cent.; N_2, 79·1 per cent. by volume.—L.U.

Ans.—

Constituent.	Vol. in 1 cu. ft.	Oxygen required cu. ft.	Products of combustion.	
			CO_2	H_2O
H_2	0·14	0·14×0·5=0·07	—	0·14
CH_4	0·02	0·02×2·0=0·04	0·02	0·04
CO	0·22	0·22×0·5=0·11	0·22	—
		0·22	0·24	0·18

Vol. of O_2 required=0·22−0·02 (the O_2 already present in the producer gas)
=0·20 cu. ft.

∴ Vol. of air required for perfect combustion

$$=\frac{0\cdot20}{0\cdot209}=0\cdot957\text{ cu. ft.}$$

Since 40 per cent. excess air is supplied

Total air=1·4×0·957=1·340 cu. ft.
Vol. of N_2 supplied with air=0·791×1·340=1·060 cu. ft.
Vol. of excess O_2 supplied=0·4×0·2 =0·08 cu. ft.

∴ Dry products are:

CO_2	0·24+0·05=0·29	14·6 per cent.
O_2	0·08	4·0 per cent.
N_2	1·060+0·55=1·61	81·4 per cent.
	1·98	100·0 per cent.

8. A producer gas has the following percentage analysis by volume: H_2, 16; CO, 20; CO_2, 6; N_2, 58.

Determine (*a*) its calorific value per cu. ft. at S.T.P., (*b*) the minimum amount of air for complete combustion, (*c*) the volumetric analysis of the products if combustion is complete.—L.U.

Ans.—(*a*) 114·9 B.Th.U. per cu. ft.; (*b*) 0·857 cu. ft.;
(*c*) CO_2, 15·5 per cent.; H_2O, 9·5 per cent.; N_2, 75 per cent.

9. A Russian crude oil has the following composition by weight: carbon, 86·9 per cent.; hydrogen, 13·1 per cent.

Determine the calorific value of the oil, the minimum weight of air required for the complete combustion of 1 lb. of the oil, and the volumetric composition of the products of combustion.

Ans.—19,480 B.Th.U. per lb.; 14·62 lb.; Products of combustion: CO_2, 13·56 per cent.; H_2O, 12·27 per cent.; N_2, 74·17 per cent.

10. A petrol has the following composition by weight: carbon, 85·2 per cent.; hydrogen, 14·8 per cent. Determine the calorific value of the oil, the minimum weight of air required per lb. of oil, and the percentage composition of the products of combustion.

Ans.—20,080 B.Th.U. per lb.; 15·03 lb. Products of combustion: CO_2, 12·9 per cent.; H_2O, 13·4 per cent.; N_2, 73·7 per cent.

11. Calculate the percentage change of volume which takes place during combustion, using the particulars given in question 7.

Ans.—7·7 per cent. decrease.

12. An engine uses oil having the following composition by weight: C, 86 per cent.; H_2, 13 per cent.; O_2, 1 per cent.

Calculate the minimum weight of air required. If 50 per cent. excess air is supplied, what is the percentage composition of the products, and the percentage change of volume during combustion.

Ans.—14·44 lb. Volumetric composition of products: CO_2, 9·3 per cent.; H_2O, 8·4 per cent.; O_2, 6·7 per cent.; N_2, 75·6 per cent.; increase, 4·4 per cent.

13. With what experimental methods are you acquainted for determining the air to fuel ratio (by weight or volume) of the working mixture of an internal-combustion engine? Describe these, and indicate the relative advantages and reliability of each method. Indicate the importance of making this determination when carrying out an engine test.—M.U.

14. The following data is taken from a paper on a producer-gas plant by Denny and Knibbs, Proc. Inst. Mech. Eng., January, 1922.

Analysis of Mond gas from producers:

CO_2, 14·9 per cent.; CO, 10 per cent.; H_2, 24·2 per cent.; CH_4, 3·1 per cent.; N_2, 47·8 per cent.

The L.C.V. of the gas at 15° C. is 126 B.Th.U. per cu. ft. An analysis of the exhaust gases gave:

CO_2, 15·2 per cent.; O_2, 2·9 per cent.; N_2, 81·9 per cent.

Determine: (*a*) theoretical ratio of air to fuel; (*b*) ratio of gas to air; (*c*) mixture strength.

Ans.—(*a*) 1·11 to 1 (by volume); (*b*) 1 to 1·37 (by volume); (*c*) 53 B.Th.U. per cu. ft.

15. The following figures are taken from the report of a trial of a four-cycle oil engine.

Analysis of dry exhaust gases, by volume, per cent.:

CO_2, 12·1; CO, 2·0; O_2, 2·0; N_2, 83·9.

Composition of the oil by weight per lb. (approx.):

C, 0·86 lb.; H_2, 0·14 lb.

The temperature of the exhaust gases leaving the engine was 800° F. (426·5° C.), and the temperature of the air 62° F. (16·5° C.).

Calculate the heat carried away by the exhaust gases per lb. of oil used. The steam formed by the combustion of the hydrogen passes away as superheated steam.

Specific heats of the gases per lb. weight:

CO_2, 0·216; CO, 0·248; O_2, 0·218; N_2, 0·244.—L.U.

Ans.—3,130 B.Th.U. (1,740 C.H.U.).

16. The exhaust gases of a Diesel engine left at 390° C., and were found to have the following composition by volume: CO_2, 9·70; CO, 0·04; O_2, 7·92; N_2, 82·34. The percentage composition by weight of the oil used was C, 86·46; H_2, 12·81. The I.H.P. of the engine was 58, and 20·6 lb. of oil were used per hour. Estimate the heat given to indicated work, and the heat carried away by the exhaust gases per lb. of oil. Take the room temperature as 15° C. and the specific heat of the gases as 0·266.—L.U. *Ans.*—3,980 C.H.U.; 2,319 C.H.U.

INDEX

[*The numbers refer to pages.*]

A

ABSOLUTE zero of temperature, value of, 64, 67
Acland, F. D., paper on Still engine, 7, 56
Adiabatic, defined, 125
  curves with varying specific heats, table of, 139
  law with constant specific heats, 73
  law with varying specific heats, 136
  relations of a perfect gas, 73, 74
After-burning, in Diesel engines, 168
  discussed, 176
Air, gas constant for, in various units, 64
  specific weight, specific volume, molecular weight, 63
  compressor, on Diesel engine, 46, 200
Airless injection engine, works on D.C. cycle, 7
Air standard efficiency, 88
Air-to-fuel ratio of Lenoir engine, 10
  of Otto silent engine, 18
  of Clerk engine, 21
  of gas engine, 149
  of Priestman oil engine, 40
  of solid-injection engine, 143
  how calculated for gaseous fuels, 149
  for liquid fuels, 154
  of petrol engine, 155
  tables for light fuel oils, 157
Akroyd-Stuart, H., first oil engine, 3, 34, 35, 36
  described, 41
Alcohol, composition of, 156
  average specific gravity and calorific value, 157
  the fuel of the future, 156
Alexander, W., entropy chart by, 129, 131
  applies Diesel card to entropy chart, 175
Alignment charts for gases, 131
Aluminium, thermal conductivity of, 180
  coefficient of linear expansion, 190
Apparent specific heats, 113, 120
Armengaud gas turbine cycle, 205
Arschaouloff system of solid injection, 172
Atkinson "cycle" engine, 22
  thermal efficiency of, 89
Atmospheric pressure, standard, 68
Avogadro's hypothesis, 111

B

Baker, H. W., and Gibson, on air-to-petrol ratios, 155
  on high M.E.Ps., 199
Balof, A., on Diesel engine piston tops, 168
Barsanti, E., and Matteucci, free-piston engine, 12
Beau de Bochas, A., brochure on his gas-engine cycle, 4
Benzol, composition of, 156
  properties of, 157
  compression ratio for, 148
Bird, A. L., book on "Oil Engines," 148
Bjerrum, determinations of specific heats, 115
Blast air, use of, in Diesel engine, 167
Blast furnace gas, properties of, 153
  compression ratio for, 148
Boyle's Law, defined, 59
Brass, coefficient of linear expansion, 190
Brayton engine, 16
  cycle of, 85
B.A. mixture, composition of, 114
  internal energy of, 122
Brons oil engine, 170
Burstall, F. W., spray-injection tests, 193

C

Callendar, H. L., on entropy, 125
  and Nicolson, thermal conductivity of iron, 179
Calorific value of gases, 153
  of light fuel oils, 157
  of crude oil, 143
  Inchley's formula for, 157
Capper, O. S., tests on Hornsby engine, 43
Carbon monoxide, properties of, 63
  dioxide, properties of, 63

Carbon dioxide, specific heat and internal energy of, 118
Carbon deposit, effect of water injection on, 197
Carburettor, in oil-engine classification, 37
  theory of, 162
  value of, as atomiser, 164
  as governing agent, 159
Carnot, S., on entropy, 125
Carnot cycle, to determine absolute temperature scale, 66
  proposed use of, by Diesel, 6, 43, 198
Carpenter, H. C., and Rugan, on growth of C.I., 178, 191
  suitable C.I. mixture for high temperatures, 192
Cast iron, thermal conductivity, 180
  coefficient of linear expansion, 190
  growth at high temperatures, 191
  suitable mixtures of, for high temperatures, 192
Charles's Law, defined, 60
Chingford, Humphrey pumps at, 30
Choke tube, carburettor, 163
Chorlton, A. E. L., on double-acting two-stroke engines, 50
Clark, H. Ade, first English tests on Diesel, 36
Classification of gas engines, 9
  oil engines, 35
Clerk, Sir Dugald, inventor of two-stroke cycle, 5
  on specification drawings, 9
  tests Lenoir engine, 11
  tests Otto and Langen engine, 11
  tests first motor-car, 6
  compressed-gas engine, 35
  super-compression engine, 202
  on the Brayton engine cycle, 85
  on composition of B.A. mixture, 114
  "zig-zag" experiments, 121
  internal energy curve, 122
  on quality governing, 158
  on carburettor types, 162
  on flame inside cylinder, 185
Clerk cycle, defined, 5
  two-stroke engine described, 19
    cycle of, 88
Coal dust, as fuel for I.C. engines, 43, 198
Coefficient of linear expansion of metals, 190
Coefficient of performance, defined, 79, 109
Coke-oven gas, properties of, 153
  compression ratio for, 148
Coker, E. G., and Scoble, on temperature changes in gas engine, 185
Cold-starting oil engine, 7, 54
Combustion chamber, shape of, in Diesel, 168
    in solid-injection engine, 174
Combustion factor, 176
  efficiency and water injection, 171
Compound steam engine used in Still system, 58
Compressed gas, use of in I.C. engines, 35, 199
Compression ratios, factors governing, 147
  table of, 148
  raised by water injection, 193
  and air standard efficiency, 88
  of Atkinson engine, 24, 90
  of Clerk engine, 21
  of Diesel engine, 48, 93, 176
  of D.C. cycle with D.C. exhaust, 97
  of Hornsby-Akroyd engine, 43
  of Humphrey gas pump, 92
  of Junkers' Diesel, 50, 169
  of modern gas engines, 24, 88, 148, 185
  of Otto silent engine, 18, 88
  of petrol engine, 148, 155, 199
  of Priestman engine, 40
  of solid-injection engine, 95
Compression of a perfect gas, 72
Conductivity, thermal, defined, 179
  of steel, 179
  of iron, 180
  of aluminium alloys, 180
  of mineral oil, 184
  of carbon, 184
Constant volume cycle on Walker chart, 141
Consumptions of Atkinson engine, 24
  of Clerk engine, 21
  of Diesel engine, 47, 169
  of Hornsby-Akroyd engine, 43
  of ideal cycles, 109
  of Lenoir engine, 10
  of modern gas engine, 25
  of Otto and Langen engine, 12
  of Otto silent engine, 19
  of Priestman engine, 40
  solid-injection engine, 56, 172
  Still engine, 58
Continental atmosphere, 69
Control of petrol engine, 159
Conversion of specific heat relations, 112, 124
  conductivity units, 179
  radiant heat constant, 187
Cook, F. J., suitable cast-iron mixture for high temperatures, 192
Coventry Quadrant motor-bicycle, 3
Crank-case compression, 48, 52
Crowther, O. H., and Lea, on temperature, effects on metals, 191
"Cut-off ratio," 93

D

Daimler, G., pioneer of petrol engines, 5
  first engine described, 40

Dalby, W. E., test of small Diesel, 176
  paper on heat transmission, 182
  conversion of radiant heat constant, 187
David, W. T., on after-burning, 176
  on radiation effects, 188
Davy safety-lamp principle applied to gas-engine ignition, 16
Denaturants in alcohol, 156
Denny, W., paper on marine engines, 56
Detonation defined, 147
Development, possibilities of, 198
Diederichs, H., translates Güldner's book, 9
Difficulties in design, 1
Diesel, R., early developments of his engine, 6, 34, 198
  his "Rational Heat Motor," 43
  engine, in nomenclature report, 36
    described, 43
    first experimental, 43
    thermal efficiency of, 92
    formation of working substance in, 167
    fuel valve for, 167
    consumptions, 47, 169
    compression ratio, 48, 93, 176
    combustion chamber, shape of, 168
    double-acting, 50
    cycle with varying specific heats, 99
      with constant specific heats, 92
      on Walker chart, 143
      on entropy chart, 175
Diesel Engine Users' Association formed, 44
Dixon, H. B., specific heat experiments, 116
  internal energy curves from his values, 122
Double-acting Diesel engine, 50
Double-port scavenge, two-stroke Diesels, 49
Dual-combustion cycle, defined, 7
  Clerk engine which worked on, 35
  thermal efficiency, 94
  with dual-combustion exhaust, 97
  with variable specific heats, 97
  on Walker chart, 143
Dulong's formula for air to fuel ratio, 154

E

Efficiency ratio, defined, 79, 109
Energy curves of gases, 118
  gaseous mixtures, 122
Entropy defined, 125
  equations, 126, 127
  chart, construction of, 133
    example of application on, 175
Ewing, Sir Alfred, on perfect and ideal gas, 72
  specific heat, conversion factor, 113
Ewing, Sir Alfred, on energy curve of gaseous mixture, 119
Examples, 210
Excess air, calculation of, 150
Exhaust gas, turbine, Rateau's, 8, 199
    analysis, 150
Expansion of a perfect gas, 72
    the working substance, 174
External-combustion engines, 145
Extraction steam engine, 200

F

Flame ports in Otto and Langen engine, 14
    silent engine, 18
    Clerk engine, 20
Float, carburettor, 163
Formation of mixtures, 1, 3
    theory of, 159
    requiring extraneous ignition, 145
    in which air only is compressed, 166
    in hot-bulb engines, 169
    in solid-injection engines, 171
    in Diesel engines, 167
Forms of internal-combustion engines, different, 4
Four-stroke cycle defined, 5
Free-piston engine, 12
Fuel valve of Diesel engine, 167
    hot-bulb engine, 170
    solid injection engine, 172
Funck, G., on supercharging, 203

G

Gas constant defined, 62
  of air in different units, 64, 65
  universal, 63, 112
Gas constants of air, CO, CO_2, H_2, N_2, O_2, 63
Gas driven motor-cars, 5
Gas engine types, described—
  Atkinson, 22
  Brayton, 16
  Clerk, 19
  Daimler, 40
  Hugon, 11
  Humphrey pump, 29
  Koerting, 29
  Lenoir, 10
  Modern, 24
  National, 28
  Nuremburg, 28
  Oechelhauser, 29
  Otto, 16
  Otto and Langen, 12
  Simon, 16
Gas thermometer, 65
Gas turbine, 8, 203
  recent papers on, 204
Gaseous Explosions Committee, 113
  on energy curve, 114

Gay-Lussac's Law, 60
General thermodynamical relation, 70
Gibson, A. H., and Baker on air-to-petrol ratios, 155
  on petrol engine M.E.Ps., 199
  and Walker on heat transmission, 185
Goudie, W. J., paper on Gas Turbine, 204
  entropy chart by, 129
Governing of free-piston engine, 13.
  by "hit-and-miss" in Otto engine, 18, 158
  quantity, in modern gas engine, 25, 158
  quality, in modern gas engine, 158
  effect on formation of mixture, 157
  of Diesel engine, 46
  of solid-injection engine, 56
Gravity, acceleration due to, 68, 69
Griffiths, E., thermal conductivity of aluminium alloys, 180
Grouping of oil engines, 37
Growth of cast iron, 191
Güldner, book on engine construction, 9

H

Haeder, H., book on engine construction, 9
Hatfield, Sir W. H., book on cast iron, 192
  on growth of cast iron, 192
Hawkes, C. J., oil-engine nozzle research, 172, 174
Heat reception and rejection, 75
  transfer across boiler-heating surface, 183
    in internal-combustion engine, 181
  transmission, 179
Helium scale in absolute temperature, 66
High-compression engine, defined, 7
Hit-and-miss governing, in Otto engine, 18
  defined, 158
Holborn and Henning, specific heat experiments, 116
Holzwarth, H., on his gas turbine, 204
  gas turbine, 206, 208
Hopkinson, B., internal energy curve, 114
  experiments with fan turbulence, 120
  experiments on heat transfer, 183
  on temperature stresses, 190
  on water-injection tests, 197
Hornsby-Akroyd oil engine, 34, 169
  described, 41
  in nomenclature report, 36
Hot-bulb in oil engine classification, 37
  engine, described, 52
    formation of mixtures in, 166
Hot plate of hot-bulb engine, 54
Hugon gas engine, 11
Human factor in design, 2
Humphrey gas pump described, 29
  cycle, 91, 97, 206
Hurst, J. E., on Diesel-engine pistons, 192
Huskisson, W. M., translates Haeder's book, 9
Hydrogen, properties of, 63
  scale in absolute temperature, 66

I

Ideal cycles, consumption of, 109
Ignition, spontaneous, temperature of, 147
Inchley, W., summarises B.A. reports, 114
  formula for calorific volume of oils, 157
Indicated horse-power, how calculated, 104
  of D.C. cycle, 106
  of C.V. cycle, 106
  of Diesel cycle, 106
Indicator cards applied to entropy chart, 134, 175
  diagram of Atkinson engine, 23
    Brayton engine, 16
    Clerk engine, 22
    Diesel engine, 47
      air compressor, 47
    Hornsby-Akroyd engine, 42
    Hot-bulb engine, 53
    Humphrey gas pump, 32
    Lenoir engine, 11
    Modern gas engine, 27
    Otto and Langen engine, 15
      silent engine, 18
    Priestman engine, 39
    Simon engine, 16
    solid-injection engine, 55
Inlet valve, velocity through, 160
Internal energy, and Joule's Law, 69
  change in, 111
  curves, derivation of, 121
  table of, for gases, 118
    for gaseous mixtures, 122
  values of, in heat units, 132
International conference of weights and measures, 68
  standards of temperature and pressure, 68
Isentropic lines defined, 125
Isobars defined, 126
Isometric lines defined, 126
Isothermal lines defined, 126

J

Joule's Law of Thermodynamics, 69
    used in entropy relation, 126
  experiments on temperature change, 70

Joule cycle, 85
  used for gas turbine, 204
Junkers' two-stroke Diesel, 49
  Diesel, combustion space of, 169
    uses super-compression, 203
    compression ratio of, 50, 169

K

Kelvin, Lord, absolute thermometric scale, 66
Kirchoff, defines a black body, 187
Koivulehto, R., and Walker, water-injection tests, 194
König, Max, paper on gas turbine, 204

L

Lamp-black, used in radiation experiments, 187
Langen, E., and Otto engine, 12
Langen, A., specific heat experiments, 115
Laudahn, W., on double-acting Diesel, 50
Launch, first petrol-driven, 6
Lea, F. C., and Crowther, on temperature effects on metals, 191
Lees, S., Deisel engine cycle with varying specific heats, 99
Lenoir gas engine, 10
  thermal efficiency of, 79
  mean effective pressure of, 103
Locomotive on Highways Act, 6
Lodge, Sir Oliver, on radiant energy, 186
Low, D. A., method of computing adiabatic curve, 138
Lubricating oil, limit of temperature with, 181
  thermal conductivity of, 184

M

Maclagen, J., double-acting Diesel, 50
Marine Oil-Engines Trial Committee, 1st report, 201
  2nd report, 58
Marine development, 46, 201
Mariotte's Law, 59
Mathot, R. E., book on engine construction, 9
Matteucci and Barsanti engine, 12
Mean effective pressures for Otto cycle, 102
  Diesel cycle, 102
  D.C. cycle, 102
    with complete expansion, 102
    with D.C. exhaust, 103
  C.P. cycle, 103
  Lenoir cycle, 103
  Otto and Langen cycle, 103
Mechanical efficiency of Diesel engine, 46, 200
  Still engine, 58
  solid-injection engine, 200
Mechanical Engineers, Institution of, on oil-engine nomenclature, 35
Mechanical-injection engine, 7
Mellanby, A. L., paper on marine oil engines, 46
Mercury, density of, 67
Mixed cycle, 7
Modern gas engine, 24
  developments of, 26
Mol, defined, 111
Molecular weights of gases, 63
Mond gas, average analysis of, 153
Moore, H., on ignition point, 147
  on denaturants in alcohol, 156
  on air-to-fuel ratios by volume, 157
Morley, A., stress due to temperature change, 190
Morley, T. B., on molecular weight of gaseous mixture, 124
  on specific heat formula, 130
  alignment chart for gases, 134
  paper on gas turbines, 204
Morrison, H. L., on American oil engines, 171
Motor bicycle, early types of, 3
    modern developments, 4
  car, first petrol-driven, 6

N

National gas engine, 28
Ner-a-car motor bicycle, 4
New cycles of operation, 201
Nicolson, J. T., and Callendar, thermal conductivity of iron, 179
Nitrogen scale for absolute temp., 66
  properties of, 63
  table of specific heats and internal-energy curve, 118
Nomogram for gaseous mixtures, 131
  chart by T. B. Morley, 134
Normal temperature and pressure, 67, 69
Nozzle of Priestman engine, 39
  of hot-bulb engines, 170
  of solid-injection engine, 172
Nuremburg gas engine, 28

O

Oechelhauser gas engine, 28, 44, 49
Oils, light fuel, properties of, 155
Oil engine, first heavy, 3
  types described—
    Brayton, 16
    Brons, 170
    Daimler, 40
    Diesel, 43

Oil-engine types described—
    double-acting Diesel, 50
    Hornsby-Akroyd, 42
    hot-bulb, 52
    Priestman, 38
    semi-Diesel, 52
    solid-injection, 53
Opposed-piston gas engine, 28
Otto, N. A., inventor of four-stroke cycle engine, 4
  cycle, defined, 5
  silent gas engine, new principles involved in, 4
    described, 16
    thermal efficiency of, 86
      with varying specific heats, 98
    mean effective pressure of, 102
Otto and Langen engine described, 12
  thermal efficiency of, 82
  mean effective pressure of, 103
Oxygen, properties of, 63

P

Paraffin, composition of, 156
  average specific gravity of, and calorific value, 157
Parsons, Sir Charles, on turbine compounding, 209
Partington, J. R., and Shilling, reference to book on specific heats of gases, 117
Pendred, L. St. L., on oil-engine nomenclature, 35
Perfect gas, defined, 59
Petrie, T., on varying specific heats, 121
  on use of Walker chart, 143
  on C.I. mixtures, 192
  on super-compression in Junkers' Diesel, 203
Petrol, composition of, 156
  properties of, 157
  and air ratio in petrol engines, 155
  engine classified, 37
    of Daimler, 40
    carburettor for, 162
Pier and Bjerrum, specific heat experiments, 115
Pipe orifices, 172
  author's experiments on, 173
Piston tops, shapes in Diesel engines, 168
  stresses in, 190
Port scavenge, two-stroke Diesels, 48
Priestman oil engine, 3, 34
  described, 38
Principles of solid injection, 54
Producer gas, average analysis of, 153
  compression ratio for, 148
Poynting and Thomson on thermal conductivity, 179
  on black radiation, 187
Pump, fuel, for Diesel, 45
  for solid injection, 55
Pye, D. R., specific heat experiments, 117

Q

Quality governing, defined, 158
Quantity governing, 25
  defined, 158

R

Radiation effects, 185
  the nature of, 186
  properties of various gases, 188
Rateau, A., exhaust gas turbine, 8, 199
Rate of heat rejection, 75
Rational heat motor, by Diesel, 6, 43
Reception and rejection of heat, 75
Reference volume line on entropy chart, 127
    table for, 130
  pressure line on entropy chart, 133
Relative efficiency, 79, 100, 109
Reynolds, O., law of heat transfer, 184
Rhead, E. L., suitable C.I. mixture for high temperatures, 192, 193
Ricardo, H., on carburettor design, 162
Richard, G., reference to Beau de Rochas, 4
Robinson, W., book on gas engines, 24
  tests Akroyd-Stuart engine, 36
Rochas, Beau de, on his cycle, 4
Rugan, H. F., and Carpenter, on growth of C.I., 178, 191

S

Sankey, Captain H. R., chairman of nomenclature committee, 35
Scavenging two-stroke Diesels, 48
Schofield, F. H., on heat conduction, 179
Schüle, W., on varying specific heat relations, 118
  on gas turbine, 204
Scoble, W. A., and Coker, on temperature changes in gas engines, 185
Scott two-stroke motor bicycle, 4
Scott-Still engine, articles on, 56
Semi-Diesel engine, development of, 6
  in nomenclature report, 36
  in oil-engine grouping, 37
  described, 52
Semi-steels for high temperatures, 193
Shilling, W. G., and Partington, reference to book on specific heat of gases, 117
Simon steam-gas engine, 16, 56
Slide valve, use of, in early types, 3
  in Lenoir engine, 10
  in Otto and Langen engine, 14

Slide valve, use of, in Otto silent engine, 17
  in Clerk engine, 20
Slow-combustion cycle of Diesel engine, 6, 167
Society of Arts, trial of Atkinson engine, 24
  of Akroyd-Stuart engine, 36
Solid-injection engine, other names for, 7
  in oil-engine grouping, 37
  described, 53
  formation of mixture in, 171
  nozzle for, 172
  combustion chamber of, 174
Solid-injection, principle of, 54
  difficulties of, 2, 174
  used in Still engine, 201
Specific gravity of Pratt's No. 1 motor spirit, 155
  of light fuel oils, 157
Specific heats, tables of, for gases, 118, 119
  of perfect gas, 59, 71
  of air, 77
  variable, 111
Specific weights of gases, 63
  volumes of gases, 63
Spray injection in Simon engine, 16
  in Hugon engine, 11
  in gas engines, 193
  in petrol engines, 197
  in hot-bulb engines, 171
Standard temperature and pressure, 67
  atmospheric pressure, 68
Steel, thermal conductivity, 179
  coefficient of linear expansion, 190
Stefan-Boltzmann law of radiation, 187
Still engine, principle of, 7
  comparison with Simon engine, 16
  described, 56
  tests on, 58, 200
Still, W. J., advocates solid-injection, 7, 201
Stodola, A., book on steam- and gas-turbines, 118
  formulæ for specific heats, 122
    used for entropy charts, 130
  on gas turbines, 203
Stuart, Akroyd, patent of 1890, 3, 169
Super-charging, 202
Super-compression, 202
  with exhaust gas turbine, 8
Summary of perfect gas relations, 76
Synthetic method of specific heats, 114

T

Temperature limits in I.C. engines, 2
  effects, 178
  stresses in metals, 189
  absolute, 65
Tensile strength, reduction due to temperature, 191
Tests of Atkinson cycle engine, 24
  of Hornsby-Akroyd engine, 43
  of pipe-orifices, 173
  of Priestman oil engine, 40
  of small Diesel, 176
  of solid-injection nozzles, 172
  of Still engine, 58, 200
  of water-injection, 194
"Text" mixture, table of specific heats and internal energy of, 119
  properties of, 123
  volume, line on entropy chart for, 131
  adiabatic curve calculated for, 139
Thermal conductivity defined, 179
    of aluminium alloys, 180
    of carbon, 184
    of cast iron, 180
    of mineral oil, 184
    of steel, 179
  efficiency, ideal, of Atkinson cycle, 89
    of Brayton cycle, 85
    of Clerk cycle, 88
    of Diesel cycle, 92, 96
    of Diesel cycle, with varying specific heats, 99
    of dual-combustion cycle, 94
    of dual-combustion cycle, with varying specific heats, 97
    of dual-combustion cycle, with dual-combustion exhaust, 97
    of dual-combustion cycle, with constant-pressure exhaust, 96
    of Humphrey pump cycle, 91, 97
    of Lenoir cycle, 79
    of Otto cycle, 86, 96
    of Otto cycle with varying specific heats, 99
    of Otto and Langen cycle, 82
Thermal efficiency, brake, of Priestman engine, 40
    of Hornsby-Akroyd engine, 43
    of Diesel engine, 47
    of gas engine with water injection, 193, 195
  new committee on, 122
  how calculated, 78
Thermodynamic scale, absolute, 66
Thermodynamics of steam engine and internal-combustion engine compared, 1
  of a perfect gas, 59
  fundamental law of, 69, 126
Thomson, Sir William (Lord Kelvin), quoted, 66
  Sir J. J., and Poynting, on thermal conductivity, 179
    on black radiation, 187
Thwaite, B. H., first user of waste gases, 26
Tizard, H. T., on detonation, 147
  and Pye, on varying specific heats, 115

Tookey, W. A., commercial tests by, 142
  gas-engine card of, 177
  on mixture strength, 148
  translates Mathot's book, 9
Tookey factor defined, 148
Total heat, change in, 111
  values of, in heat units, 132
  curve on Walker's chart, 136
Transmission of heat, 179, 182
Triple-expansion steam engine used in Still system, 58
Turbine design, 208
Turbo-compressor, Rateau's, 8, 199
Two-stroke cycle defined, 5
    thermal efficiency of, 88
  Diesel's, 48
Tyndall, J., on heat absorption of CO_2, 188

U

Uniflow steam-engine cycle, 97
Universal gas constant, 63, 112, 124
Unwin, W. C., tests Priestman engine, 40

V

Valve scavenge, two-stroke Diesel's, 48
Vaporiser in oil-engine grouping, 37
  in Priestman engine, 39
Variable specific heats, effect on thermal efficiency, 97
  and energy curves, 111
  units of, 111
  of gases, 118
  of gaseous mixture, 119
Volumetric efficiency, correction on Walker chart, 142
  and Tookey factor, 149
V-type twin petrol engine, first, 6

W

Walker, W. J., on D.C. cycle, 7
  on various forms of D.C. cycle, 96, 97, 98
Walker, W. J., on apparent specific heats, 121
  internal energy curve of, 122
  chart, construction of, 135
    use of, 141, 143
    for detecting after burning, 177
  adiabatic curve of, 139
  and Gibson, on heat transmission, 185
  and Koivulehto, water-injection tests of, 194
Waste heat boiler and exhaust gases, 200
Water injection in hot-bulb engines, 171
  to raise compression, 193
  tests of Hopkinson, 197
    of Burstall, 193
    of Walker and Koivulehto, 194
  vapour, table of specific heats and internal energy of, 118
Watson, W., on air-to-petrol ratio, 155
  on fuels for I.C. engines, 156
Weisshaar, E., on after burning in Diesels, 176
Wells, G. J., on thermal efficiency, 122
Wimperis, H. E., method of computing adiabatic curve, 139
  on carburettor theory, 163
  on cooling piston-rim, 182
  on Otto cycle with varying specific heats, 97
Work done by a perfect gas, 74, 208, 209
Working substance, present knowledge of formation of, 1

Y

Yarrow, Sir Alfred, responsibility for introduction of Locomotives on Highways Act, 6

Z

Zig-zag experiments of Sir Dugald Clerk, 121

THE END

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